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ENGINEERING LABORATORY, INC.
MIDDLETOWN, CONNECTICUT

ENGINEERING REPORT

SUBJECT: The Analysis, Modification & Test of a Torsional Hairspring Escapement

CLIENT: Frankford Arsenal, Phila. 37. Pa.
To: Armed Services Technical Information Agency
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FINAL REPORT PHASE I

THE ANALYSIS, MODIFICATION AND TEST OF A TORSIONAL HAIRSPRING ESCAPEMENT

Reference (a) Contract No. DA-36-038-ORD-21198-M
(b) V.E.M. No. FA708.1 dated January 19, 1961 "Summary of the Calculations On Frankford Escapement Model"

(REL W.O. 708)

for

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August 11, 1961

Final Report Phase I
REL Report No. 593-9
REL W.O. 708

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REL Report No. 593-9
REL W.O. No. 708

TABLE OF CONTENTS

Table Of Contents, Continued

REL Report No. 593-9
REL W.O. 708

Copy No._____

FINAL REPORT PHASE I

THE ANALYSIS, MODIFICATION AND TEST OF A TORSIONAL HAIRSPRING ESCAPEMENT

Contract No. DA-36-038-ORD-21198-M Dated October 18, 1960

Reference: REL Technical Proposal Dated August 9, 1960

1.0 SUMMARY OF PHASE I

The purpose of this program has been the analysis, modification and testing of a Frankford Arsenal designed torsional hairspring escapement mechanism for use as the governing mechanism in a spring driven gear train. The mechanism was to be considered accurate if the lever frequency variation did not exceed $\pm 0.2\%$.

The following report summarizes Phase I of this program in which the analysis and modifications were completed. The report consists of a theoretical discussion of this type of escapement, an outline of the approach taken to solve the difficulties it presented, and a report on the results of the studies made. In paragraph 3.4 a summary of the changes made to the Frankford design based on this work is given.

Testing done during Phase I was restricted to bench tests, the results of which are given in paragraph 4.0. As a result of these tests it may be stated that the escapement does have the required accuracy under these conditions when the recommended changes are incorporated.

Phase II, which will consist of the extensive spin testing of a sample lot of thirty (30) units is to follow upon the acceptance of this report.

2.0 GENERAL DISCUSSION

2.1 Introduction

Every mechanical timekeeper consists of three basic elements:

1. POWER SOURCE - Early clocks were powered exclusively by falling weights; with the introduction of springs, transportable clocks, and later, watches, were made possible.
2. GEAR TRAIN - Driven by the power source, the gear train links the power source and the escapement in the proper ratio and provides the desired output.
3. ESCAPEMENT - The governing system to accurately control the rate of motion.

The third of these, the escapement, is composed in turn of three components:

1. SCAPE WHEEL - A gear whose tooth shape is designed to allow indexing by the pallet, and also to provide power through the pallet to the balance system.

2. PALLET - This is essentially a ratchet for alternately engaging and releasing the teeth of the scape wheel. The rate of oscillation of the pallet is determined by the balance system, but the scape wheel, when moving, supplies power to the balance through a camming action against the pallet. Thus the pallet is driven alternately by the balance and the scape wheel.

Pallets have taken countless forms and many names: cylinder, virgule, detent, etc., the name frequently identifying the type of escapement. Properly, the term pallet refers only to the impulse faces with which the teeth of the scape wheel engage. More generally the whole piece on which these faces are formed is called the pallet. In this study, the pallets are designated as the impulse faces of the cylinder.

3. BALANCE SYSTEM - Basically an oscillating or vibrating system which provides the timing capability by regulating the motion of the pallet.

This is the brain of every timepiece, for on the accuracy of the balance depends the usefulness of the whole device. In fact, every major advance in mechanical timers, with the exception of the introduction of the mainspring, has followed an improvement in the type or use of the balance. This is more fully discussed in the succeeding sections.

2.2 Historical Considerations

The use of the cylinder escapement as a governor for clocks and watches spanned a 200 year period between 1700 and 1900. It replaced the earlier recoil, or verge escapement, and was itself replaced by the detached lever. The periods during wh_ch these three escapements developed, were extensively used, and declined are roughly shown in Figure #1.

The superiority of each of these escapements over its predecessor was almost exclusively due, as mentioned in the last section, to major improvements in the balance system. In the earliest escapement the verge, the balance and the pallet were combined on a common arbor, or shaft. Since hairsprings were unknown, the resulting inertia system was never free of the scape wheel, and had to be driven in both directions. In addition, the scape wheel had to stop the balance and reverse its rotation, thus causing a momentary reversal of the scape wheel and, of course, the entire gear train; hence the term "recoil". (See Figure #2)

The verge escapement was designed to be powered with falling weights, for mainsprings were not yet known, and this constant force drive made its use feasible. The verge is useless as a timekeeper when driven by a spring unless fitted with a stackfreed or fusee to equalize the spring force for its rate is proportional to the applied force.

With the introduction of the pendulum for clocks and the balance spring for watches it was possible to have a balance that would swing of its own accord, rather than being pushed by the scape wheel. Shortly after the pendulum came into use, George Graham invented his famous "dead-beat" escapement. The design of this is such that no recoil or reversal of the gear train resulted, hence the name dead-beat. It is not, however, a "free" escapement, for the pendulum balance is never free from the escape wheel. When the pendulum is not being impulsed, a tooth of the scape wheel is locked against one of the moving pallets and the friction there affords a drag on the pendulum. A sketch of this escapement is shown in Figure #2.

If the arms of the pallet of the dead-beat escapement are made shorter the pallet arbor must be brought closer to the wheel and fewer wheel teeth will be included between the pallet faces. By making the pallets so short that they embrace only one tooth, the Graham horizontal or cylinder escapement results, also shown in Figure #2. By replacing the pendulum with a hairspring and balance wheel, this cylinder escapement presented the first serious competition to the verge. By the middle of the nineteenth century the Swiss had so developed this escapement and its production that the thin, light, cheap and yet reliable watches they made almost drove all others out of the market.

As has been mentioned however, this was not a "free" escapement. When the balance is not being impulsed, there is a tooth of the scape wheel resting against the cylinder. The friction there is not negligible as it is in a large clock, for the balance does not have the great momentum that a heavy pendulum has. Frequent oiling to reduce friction was always necessary with this escapement.

Also, when applied to watches, the arc of travel had to be greatly increased as it was not possible to make such an escapement work well with the usual small pendulum arc. For these reasons it was not until electricity was employed in connection with clocks that any free escapement for pendulum clocks was attempted, but the cylinder escapement for watches was soon replaced by the detached lever.

The detached lever escapement is a variation of the Graham dead-beat as is the cylinder, but with a "free" balance. By separating the balance system from the pallets the balance is allowed to travel freely during most of its oscillation and relatively uninfluenced by extraneous forces, is capable of far greater accuracy. In fact the detached lever escapement has made possible tiny watches that approach in timekeeping ability, the performance of large marine chronometers. A modern detached lever escapement is shown in Figure #3. This survey would not be complete without mentioning two relatively recent developments in the timer field.

The first is the introduction of the electric and electronic watches. As the electric clock has largely supplanted all others, it may be that the detached lever escapement, now almost universally used for watches, is, like the verge and the cylinder, to be replaced.

The second is that the great increase in the use of military timers has revived, in various forms, the use of almost forgotten escapements. Hence the verge, the cylinder, and the detached lever are all to be found in modern military devices. This will probably continue to be so since each has useful features that are not inherent in the others.

3.0 STUDY OF THE TAVERO ESCAPEMENT

This escapement, a modern version of the once popular cylinder escapement, was developed and manufactured by, and received its name from Tavero S. A., a Swiss firm located in Geneva. Several million fuzes utilizing the escapement were produced for the British and other governments, and at various times the United States has been interested in its performance. The present version, modified by Frankford Arsenal, experienced difficulties of operation that led to this study, undertaken by Raymond Engineering Laboratory on October 19, 1960.

3.1 Purpose of Study

The purpose of this program is to analize, modify and test the Frankford Arsenal torsional hairspring cylinder escapement for use as a governing mechanism of a spring driven gear train. The program is divided into two phases, Phase I consisting of an analysis and revision of the Frankford design, and Phase II, extensive testing of the revised design. This report concludes Phase I.

3.2 Outline of Study

Phase I was started with an examination of the original design and a review of the development carried out by Frankford Arsenal. It was understood at that time that not only did the escapement fail to meet the required specifications, but that its basic operation was faulty. Based on this initial investigation, therefore, two objectives were outlined as guides in incorporating revisions to the design,

1. The primary goal must be the assurance of basic operation.
2. Additionally, the escapement should be optimized from the standpoints of accuracy, reliability and manufacture.

To accomplish these objectives, several different avenues of approach were taken. These are briefly described below.

3.2.1 Historical Study

A study was made of the history of the cylinder escapement. This is summarized in section 2.1 of this report. From this came the assurance that the principle was sound, despite the rather significant differences in geometry between the historical and the present designs, as shown in Figure #4. In addition, this work provided knowledge of the inherent advantages and disadvantages in this type of escapement, and suggested areas to be investigated and manufacturing techniques to be considered.

3.2.2 Full Size Models

Several full size models have been fabricated during Phase I to confirm revisions made to the design. The first of these models duplicated as nearly as possible the original design, and its failure to operate properly was in line with Frankford Arsenal's experiences. The final model has incorporated all the revisions made and is a prototype for the fabrication of a test lot of units to be evaluated in Phase II. Testing of these models was checked on a 98.772 beat

Recording Watchmaster, and the escapements were all run on a precision 80:1 gear train with falling weight drive.

3.2.3 Large Scale Model

Because of the difficulties inherent in working with tiny mechanisms, a large (thirty times full size) working model of the escapement was constructed. This model was both geometrically and dynamically similar to the Frankford design, but operated at the much slower rate of about one cycle per second. Such parameters as scape wheel inertia, balance inertia, spring constant, mainspring torque and scape wheel to cylinder centerline distance were adjustable, and redesigned parts could be quickly and inexpensively made and tested.

This device has been an invaluable development tool at all stages of the study. When first tested, it was set up to duplicate the original design and, as with the first full size model, was seen to operate improperly. It was immediately apparent that the spacing between scape wheel and pallet was too great, causing operation akin to the verge or runaway escapement.

This was quickly corrected and the observation confirmed.

In addition to simulating all the changes in the above parameters recorded in the Frankford Arsenal study, this model was used for the following studies:

1. Impact patterns were displayed for study through an acoustical pickup on an oscilloscope.
2. Movies taken of the model were used in studying the impulse phase of the cycle.
3. A number of pallet and one alternate scape wheel configurations were made and tested.
4. The arc of balance oscillation under ideal conditions was determined.
5. Operation duplicating Frankford Arsenal tests was simulated to further confirm the revisions made to the original design.

A full report on the design and use of this device is contained in a report (reference "B").

3.3 Detailed Discussion

3.3.1 Theory of the Cylinder Escapement

In a timing device such as the Frankford Arsenal escapement, we have, first of all, a power unit consisting of a spring-driven gear train. It is intended that this unit perform some simple switching function following a prescribed lapse of time. The timing requirement necessitates some means of controlling the turning rate of the powered gears. For low power applications, in which precision timing is of prime importance, the basic power unit must be supplemented by a constant-rate control device.

An "ideal" torsion-balance pendulum, if undisturbed, is such a constant-rate device. However, when we attempt to use a pendulum to influence and control the basic power unit, the latter must necessarily react in reciprocal fashion upon the pendulum. When the ideal pendulum is subjected to external influence (or disturbance), the natural period of the pendulum tends to alter.

Thus, the Frankford torsion balance is disturbed to a greater or less extent by normal bearing frictions, by scapewheel tooth drag on the inner and outer cylindrical pallet faces during each dead-beat phase, and by some sort of positive torque pattern during each impulse phase. To begin with, the balance falls short of the ideal pendulum due to ever-present bearing friction. When used to control power flow in an escapement, drag friction is inevitably introduced as an additional external disturbance. An impulse or positive torque phase is then necessary to restore energy lost in friction which would otherwise cause continuous reduction in amplitude. Though it is theoretically possible to balance this pattern of positive and negative disturbances so as to maintain the ideal or free-pendulum period, in practice such a critical balance can be approached but not absolutely achieved. Hence, the natural timing cycle is altered somewhat.

This fact alone could not explain a continuous timing "drift", if the degree of deviation from the optimum condition were constant for all real time. We are lead to the rather sobering realization that regardless of how successful we might be in efforts to include all deviating factors in an exact "paper" analysis, the resulting balance cycle-rate could theoretically be calibrated in real time and constant-rate control and consistent long-term repeatability would be expected. If, in practice, the so-called beat-rate appears to wander about some mean value, it must be concluded that some pertinent parameter, rather than remaining constant, is shifting so as to modulate the basic beat frequency. Similarly, a continuous unidirectional shift in rate must be indicative of parameter drift. The point is that neither an idealized analysis nor even one which considered the detailed complexity of deviations from the optimum would predict a varying control rate. Such rate variations must arise from "variation of the deviations"; in other words - a third sub-order effect. A fluctuating friction coefficient is one example we might reasonably expect to encounter. Beat frequency fluctuation could thus be explained, though short-term frequency drift is less easily attributed to friction.

It would seem, then, that we might optimize our tuned escapement by striving both to minimize deviations from the ideal "free balance" and to balance out possible parameter fluctuation tendencies. This conslusion appears to put us back where we started. However, it is important to note the subtle distinction between "deviation" and "variation", as previously discussed. We need only be concerned with deviations (external disturbance of the ideal rate-controlling balance) insofar as time-variation thereof may result in a varying pendulum period, which in turn would reflect as a varying escape rate of the power mechanism.

To restate and expand this in slightly different form, the accuracy of any escapement will be largely determined by the ability of the hairspring and inertial balance system to behave (as closely as possible) in accordance with the laws governing a free pendulum. The factors acting to thwart this ideal are:

1. Disturbances caused by the scape wheel. This is the major problem.
2. Bearing friction, which is reduced as much as possible.
3. Windage, which may be neglected.

The first of these, the scape wheel disturbances, are of three distinct varieties, corresponding to the three phases of the escapement cycle:

1. The impulse phase, during which, ideally, the scape wheel should impart just enough energy to restore that dissipated in frictional and other losses in the balance system while causing the least disturbance in the period of that system. To accomplish this, the required energy should be transferred at the point of maximum velocity in the balance travel; i.e., the center of the total amplitude. Ideally the pulse is instantaneous. Actually it should be distributed over the shortest possible time.
2. The "dead" phase, during which the scape wheel is stopped. In the cylinder type of escapement, the scape wheel tooth remains in contact with the rotating

balance system, resulting in additional friction forces. (The fact that the balance of a detached lever escapement "breaks free" of the scape wheel after one impulse, and completes its travel relatively free of all but pivot forces accounts for the inherently greater accuracy of this type of escapement.)

3. The "free" phase, beginning when the scape wheel tooth leaves the pallet face following an impulse, and ending when it is again stopped against the cylinder. The effect on the balance system of the impact and the "bouncing" of the tooth is uncertain, tending to complicate analysis of the dead phase friction forces.

Bouncing, like all the other factors tending to cause deviations from the ideal "free balance", should not, in itself, affect the time-keeping abilities of an escapement. But this is the type of factor whose effects may vary with time, and which should therefore be minimized from that standpoint.

3.3.2 Design of the Cylinder Escapement

The design of a cylinder escapement can be defined by determining the following variables:

Scape Wheel

1. Diameter
2. Number of teeth
3. Length of tooth
4. "Lift" of tooth impulse face
5. Shape of tooth impulse face
6. Effective inertia
7. Material
8. Reflected mainspring torque

Balance System

9. Cylinder-scape wheel spacing
10. Shape of pallet faces
11. Cylinder material
12. Cylinder and balance inertia
13. Hairspring constant

When these are determined on the basis of the considerations discussed in the preceeding section, and modified by the realities of fabrication, a successful cylinder escapement should result. It will be noted that many factors are not listed above, but their determination is largely fixed by one or more of these variables. For example, the cylinder diameters are determined by the scape wheel, and the beat rate by the balance system inertia and the hairspring constant.

In the following paragraphs each of these variables is discussed as it applies to the Frankford Arsenal design, as, during Phase I, each was considered and studied.

3.3.2.1 Scape Wheel:Diameter

The diameter of the scape wheel largely determines the scale of the entire escapement and is hardly a matter of choice, since it is usual to make this as small as possible, consistent with manufacturing capability. In Table I the scape wheel diameter, along with several of the other variables, is compared with three other escapement types. These are:

1. A historical cylinder escapement shown in Figure #2.
2. A modern detached lever escapement (Hamilton design) shown in

Figures #3 and #5 (Drawing #1582-15).

3. A REL Junghans escapement shown in Figure #5 (Drawing #1582-15). This is a dead-beat escapement, but not detached, employing a flat spring colleted on the pallet-balance arbor.

It will be noted that despite the widely divergent types, the scape wheel diameters, as expected, are very nearly identical.

3.3.2.2 Scape Wheel:Number of Teeth

Also listed in Table I are the number of teeth on the scape wheels of the various escapements. It is immediately noted that the Frankford design has the fewest number of teeth by a wide margin. Furthermore, in no other reference can a scape wheel of fewer than twelve teeth be found. This factor, then, bears further investigation.

From a kinematic standpoint, of course, the number of teeth provided is entirely arbitrary. But the selection made will determine the angle through which the scape wheel rotates in each cycle; hence, for a given beat rate, the speed of that wheel. Fewer teeth will result in a faster wheel. Since each tooth of the faster wheel will lock more often and with greater impact, the factors to be considered when providing a small number of teeth are:

1. The effect of the higher impact on the timing ability of the escapement. This higher impact will result in greater "bouncing" of the scape wheel tooth against the cylinder wall. As discussed in Section 3.3.1, this bouncing will not, in itself, affect the time keeping capability, so long as the bounce effect is constant. But the vagaries of this action are such that a constant effect cannot be guaranteed - leading to the conclusion that minimizing

of the bouncing will reduce the variations in its effect.

2. The wearing of the scape wheel teeth, which affects the life of the escapement.

Because of the small number of teeth used in the Frankford scape wheel, each tooth experiences a considerably greater number of impacts per unit of real time than is usual. This is true, as shown in Table I, whether the beat rates of comparable escapements are identical or as actually used.

Despite the fact that the Frankford design may have an inherent handicap in this respect, it was decided early in the study to retain the eight-tooth wheel without change. Three reasons led to this decision.

1. To do otherwise would have meant loosing the identity of the Frankford design, and perhaps obscuring the reasons for difficulties experienced with that design.

2. Uncertainty as to the seriousness of these practical considerations.

3. Difficulties in providing alternate scape wheels.

This decision seems to have been justified, since the escapement, as revised, operates satisfactorily. However, it is felt that in an entirely new design, the use of from twelve to sixteen teeth should be seriously considered.

3.3.2.3 Scape Wheel:Length of Teeth

Determination of the number of teeth defines the tooth pitch; in the case of the Frankford design, the pitch is 45°, or an arc length of

$$\text{Wheel dia. } \times \pi \div 8 = \underline{\underline{.1298''}}$$

Since the tooth is enclosed by the cylinder inner diameter, and the space between teeth by the cylinder outer diameter, it is obvious that the length of tooth must be less than half of this arc length, and that how much less will largely determine the cylinder wall thickness.

Table I shows the comparison between the eight-tooth Frankford design and the fifteen-tooth historical escapement in this respect. The similarities are striking, particularly as to the percentage of arc length filled by the tooth, and the clearances between cylinder and scape wheel teeth. But the differences, due to the difference in number of teeth, are more important. While the percentage of arc filled by the tooth in the Frankford design is almost identical to the other, this results in a much thicker cylinder wall, and, of course, the actual tooth length is almost double that of the fifteen-tooth wheel. (See Figure #4) These factors have two important effects.

First, as mentioned in paragraph 3.3.1, the impulse phase of the cycle should be instantaneous, or, more practically, distributed over the shortest possible time.

Obviously, as the length of tooth and wall thickness of the cylinder increase, the impulse phase tends to lengthen, thus departing further from the ideal.

Secondly, with the escape wheel defined, there is an ideal spacing between cylinder and scape wheel, based on minimizing frictional effects, for any given cylinder diameter. This ideal can only be realized when the cylinder wall thickness is zero; that is, when the outside diameter equals the inside diameter. As the wall thickness increases, more of a compromise is necessary in selection of the spacing, with a corresponding increase in frictional effects. This is discussed in greater detail in paragraph 3.3.2.8.

In general, then, it may be concluded that the tooth length should be chosen to result in a cylinder of the least wall thickness. In the present design no change was made, for the reasons outlined in paragraph 3.3.2.2.

3.3.2.4 <u>Scape Wheel: "Lift" and Shape of Impulse Face</u>

Perhaps the most difficult factor to be considered in designing a cylinder escapement is the precise shape to be given the impulse face of the scape wheel, since the geometry of this and the pallet impulse faces are almost inseparable and their interaction is extremely complex. The best procedure, however, is to determine the scape wheel design first, and then fit the cylinder to it.

In this study, the approach followed, for reasons discussed above, has been to leave the scape wheel unchanged. It was considered, however, that a small change in the geometry of the impulse face could be made if such action was indicated. Accordingly, this area was studied from both historical and empirical standpoints. As a result, no changes were made.

Historically, we find impulse faces from flat to curves of complex character. The classical argument, which has considerable validity, is as follows (refer to Figure #6):

When the incline is straight (a-x-c), the "lift" of the cylinder by the scape wheel is at first small, increasing toward the end of the impulse phase. This tends to accelerate the cylinder rapidly, resulting in a larger deflection of the balance system than other shapes would provide, and at the same time decelerating the scape wheel so that it leaves the cylinder with a lesser velocity and, hence, a lighter impact.

When, on the other hand, the incline is of a curvature greater than the diameter of the scape wheel (a-b-c), the angular displacement of the tooth is at first very slight, but it increases with great rapidity. It traverses its effective path with a gradually increased acceleration, and, completing the latter portion of its lift with great rapidity, falls with violence on the cylinder rest surface. Since it is more difficult to secure a permanent rate with an escapement in which the drops are excessive, this type of curvature is to be avoided.

When the radius of curvature of the impulse face is approximately the same as that of the scape wheel, the angular movement of the wheel corresponds with that of the cylinder, and both are (roughly) uniformly accelerated. This type of curvature gives approximately the desirable results of the straight incline, but will wear less, and therefore was the most popular type in cylinder watches.

The Frankford Arsenal design, shown dotted on Figure #6, has a curvature slightly in excess of that recommended by history, but probably not so great as to be troublesome. Here another factor should be considered: that of the angular displacement of the balance system.

It has been mentioned several times that the impulse should be as brief as possible, or, to put it another way, should occupy as small a portion of the total balance amplitude as practical. One way to accomplish this is to increase the swing of the balance during the rest phase. As this principle was well known, the straight incline, imparting greater acceleration to the balance, was considered ideal. With the present design, however, a compromise must be made because of space limitations, and the greater curvature may be justified on that basis.

There is a second practical reason for the slightly excessive curvature. Having accepted a cylinder of heavy wall thickness, we are not entirely free to select the incline geometry at random. When a scape wheel with straight impulse faces on the teeth was tried on the thirty times size model, it would not function simply because of interference with the wide impulse face of the cylinder which had to be traversed. In crossing the impulse face of the cylinder, the point of contact must transfer from one cylinder diameter to the other and a certain amount of tooth curvature is necessary to effect this transfer smoothly.

The decision to recommend no change in this area was made on the basis of the above considerations, but it was felt necessary to provide some empirical supporting information. Accordingly, the thirty times model was run and photographed with the results shown on Figure #7 (Drawing #1582-D-17) The curves shown on this drawing compare the actual motion of the balance system with a computed sine wave, representing the "ideal" motion. The similarity between the actual and the ideal is striking, despite the many doubts that have been raised about the geometry.

Also shown on this drawing are curves of scape wheel rotation versus time during the impulse phase. It is immediately apparent that the angular displacement of both cylinder and scape wheel is surprisingly constant with time, and that if the impulse periods seem to extend over an appreciable portion of the entire cycle (as opposed to the "ideal" instantaneous impulse), it would be hard to realize a better overall pattern.

In conclusion, then, it is felt that in an entirely new design a larger number of teeth would be provided, and this might dictate a flatter impulse edge. But having accepted the eight-tooth design, the present configuration is entirely acceptable.

3.3.2.5 Scape Wheel:Effective Inertia

While the effective inertia of the scape wheel and gear train is an important factor, little really needs to be said about it as other design considerations usually dictate its value. The overall scale of the escapement determines the inertia fairly closely and the other variables are designed to fit.

In Table I the relative inertias of the three military escapements are given. They are all about the same, as would be expected. No further investigation of this factor is considered necessary.

3.3.2.6 Scape Wheel:Material

The stainless steel used in the Frankford Arsenal design is certainly the best choice. The corrosion resistance afforded is desirable, of course, and there seems to be no need to change to the harder carbon steels. As is shown on Table I, the various escapements use both these metals, plus brass. A considerable amount of running has been done in tests with the Frankford-supplied stainless wheels; they have all stood up well and have not caused excessive wear on the cylinders.

3.3.2.7 Scape Wheel:Mainspring Torque

The Frankford Arsenal escapement was designed to operate with a mainspring providing between 15 and 25 inch-ounces of torque. It was found through tests run on full size escapements that in this range, the beat rate will vary about 0.8%, or twice the allowable $\pm$0.2% accuracy. Below 15 inch-ounces the beat rate drops off rapidly and above about 27 inch-ounces there is a sudden discontinuity, and the beat rate increases very rapidly with an increase in applied torque. The curve of beat rate versus torque is shown (dotted) on Figure #8 (Drawing #1582-D-14).

Hence, it would appear that the unit is inherently torque sensitive, and incapable of holding tolerance under anything but constant torque.

The picture changes rather markedly, however, when the cylinder is lubricated. Under these conditions, the escapement might be capable of holding the desired 0.4% beat rate tolerance over a range of from 10 to 25 inch-ounces. This is shown by the solid upper curve of Figure #8 which shows runs made on the same escapement with a solid film lubricant coating applied to the cylinder.

The two solid curves shown indicate that there is a "hysteresis" effect when the applied load becomes great enough to overstress the mechanism. The safe torque is about 30 inch-ounces.

More will be said about lubrication in paragraph 3.3.3.2, but as far as the applied torque is concerned, it is essential to acceptable operation of the escapement.

Variation in torque is probably the greatest problem to be encountered in spin environments because of the effect of spin on the output of the mainspring. Studies of this are scheduled for Phase II of this program.

3.3.2.8 Balance System:Cylinder - Scape Wheel Spacing

The outside and inside diameters of the cylinder are determined by the scape wheel; the O.D. by the space between teeth, the I.D. by the length of the tooth. The position of the center of the resulting cylinder relative to the scape wheel centerline should be chosen to reduce the frictional drag on the cylinder to a minimum during the "dead" phase. The scape wheel drag during this phase will, of course, affect the ability of the balance-hairspring system to obey the laws of the free pendulum as discussed in paragraph 3.3.1. Were this effect constant during the entire travel of the balance, it might be neglected, but since friction forces are seldom constant, they should be reduced as much as possible so that the inevitable variations can be minimized. Ideally, the line of action of the force exerted on the cylinder by the scape wheel should pass through the center of the cylinder; this will result in the smallest possible tangential forces on the cylinder.

This, however, cannot be accomplished unless the I.D. and the O.D. of the cylinder are the same. It can be shown graphically that the ideal centerline distance for the dead phase on the O.D. is about 0.1591" and on the I.D. 0.156" (note that the originally designed 0.1626" is greater than both of these). The best compromise with the present design is the average of these; about 0.158". It is interesting to note that the first Frankford Arsenal design used 0.158 for this dimension. A detailed discussion of friction drag on the cylinder and the graphical solution which is shown on drawings #1582-4 and #1582-5 is given in Appendix A. This solution, plotted on Figure #9, confirmed earlier tests with the 30x model which indicated that the designed center distance of 0.1626" was too great, and that the tolerance would allow this to exceed 0.1640 at which the escapement would no longer operate correctly.

Accordingly, this dimension was changed to and fixed at 0.158".

3.3.2.9 Balance System:Shape of Pallet Faces

With the presently designed cylinder pallet (impulse) faces, only a small portion of those surfaces is contacted by the scape wheel tooth. This is shown in Figure #10, No. 1. The tooth actually "skips" most of the receiving (R) pallet (so-called because this side "receives" the tooth) and hardly touches the leaving (L) pallet at all. Initially, this was felt to be a very bad situation and a number of other configurations- some of which are shown on Figure #10- were tried. It was found possible to maintain contact with the tooth to a greater extent with some of these designs, and particularly with Number 5. But it was also seen, rather surprisingly, that no change in the overall action was achieved regardless of the specific design.

It was felt, consequently, that the original design should be largely retained as being more reasonable of manufacture.

The only alteration, therefore, was a slight rounding of the corner between the receiving pallet and the cylinder outer diameter, and this was primarily decided to reduce wear rather than to achieve better operation.

3.3.2.10 <u>Balance System:Cylinder Material</u>

Stainless steel, hardened to about Rockwell 35 to 40, as originally designed has proved to be entirely satisfactory. No change has therefore been made.

3.3.2.11 <u>Balance System:Hairspring Constant and Balance Inertia</u>

The design of the torsional hairspring requires only the definition of length (L) and cross section. The length to be used can be quickly chosen on the basis of available space, since a longer spring is easier to adjust to a given beat rate than is a shorter one.

For the present design $L_{max.} = 1.070"$ but to provide leeway the value <u>L = 1.000"</u> was chosen.

With the length determined, a cross section must be found which will result in the desired beat rate or frequency (F), and which can be manufactured and assembled easily.

In this design the desired rate is 100 beats per second. Then F = 50 cycles per second.

For an undisturbed or "free" torsion pendulum, frequency is a function of the spring constant (K) and the balance inertia (I):

$$F = \frac{1}{2\pi}\sqrt{\frac{12K}{I}} \qquad (1)$$

F = frequency (cycles/second)

K = spring rate (lb-in)

I = moment of inertia of the balance ($slug \cdot in^2$)

Values of F vs. K for $I \cong 5.0 \times 10^{-7}$ are plotted in Figure #11, where this value of I represents the moment of inertia of the Frankford balance lever with the adjusting screws turned out slightly. Thus, if the Frankford balance were a "free" pendulum, a frequency $F = 50$ cps. would theoretically require a torsion spring constant $K = 0.004099$ pound-inches.

During tests made with Frankford-supplied (0.010" x 0.003") hair-springs, it was found that a rate of approximately 50 cycles/second (actually 49.39 cps.) was achieved with a spring length of about 3/16". It is possible to calculate a spring rate on the basis of these dimensions:

$$K = \frac{\dfrac{b^3h^3G}{(b^2+h^2)}}{KL} \qquad (2)$$

where b and h are the cross section dimensions (inches)

G is the modulus (9.4×10^6 for Ni-Span-C)

k is a factor dependent on the ratio h/b.

For b = 0.003", h = 0.010", L $\cong$ 3/16", h/b = 3.33 and k = 3.41

$\therefore$ K = 0.00364 lb-in.

This value of K is somewhat less than that which the "free pendulum" curve (see #10, Figure #11) would predict, which indicates the degree of interference (impulse and friction) experienced by the balance assembly. A new cross section was therefore chosen to give approximately the same value of K, using a new spring of length L = 1.000".

(Various Frankford-measured frequencies also appear as discrete points in Figure #11, in which the various spring dimensions utilized are as indicated in the included table. Two additional free-pendulum curves have been plotted, for $I = 3 \times 10^{-7}$, and $I = 4 \times 10^{-7}$. If the Frankford test data were known to be correct and accurate, we might interpret the rather substantial deviation of the test points from an assumed free-pendulum curve as being indicative of complex variations in total disturbance due to the combined influence of friction and impulse on the motion of the torsion balance. However, it does not seem wise to attach too much significance to these "carried-over" test results.)

In order to obtain approximately 50 cycles/second with a longer torsion spring, the approximate previous value of K was maintained by means of changing the spring cross section. Letting $1/B = K(1+b^2/h^2)$, equation (2) becomes:

$$KL = Bhb^3G \qquad (3)$$

The product KL depends only on h and b for a given material. Furthermore, since $K \sim 0.00364$ lb-in.. and $L \sim 1.000$, any combination of h and b must satisfy equation (3) for a constant KL $\sim$ 0.00364 lb-$\overline{\text{in}}^2$.

Substituting KL = 0.00368 (note slight arbitrary change) and G = 9.4×10^6 in equation (3), and solving for various values of the ratio h/b, gives the combination of h and b plotted in Figure #12. Any combination of h and b which lies on this curve of KL = 0.00368 should result in a spring again producing approximately 50 cycles per second at a new spring length of about 1.000".

Figure #12 shows part of a curve which is one of a family of constant KL curves. Several other values of KL have been plotted on Figure #13. It is apparent from inspection of Figure #13 that, at higher values of h/b, the smaller dimension becomes increasingly critical. Uniform manufacture of a spring with a high h/b is thus more difficult, and a greater range of adjustment of L would have to be provided to maintain the desired K.

The lower limit of h/b is 1.0; that is, a square spring. Unfortunately, short of expensive diamond dies or other exotic manufacturing techniques, such springs are impossible to manufacture, since rectangular spring wire is made by rolling round wire through single rollers; side rollers are not used. h/b values close to 1.0 are worth considering. By arbitrarily selecting $h/b \cong 3.0$ as an upper limit, the following wires were selected from Figure #12 for trial.

.014" x .00475" $h/b = 2.95$

.010" x .00570" $h/b = 1.76$

.008" x .00650" $h/b = 1.23$

The edges of rectangular wire retain a curved shape due to the manufacturing technique, hence the above samples are not truly rectangular. The last, in fact, is little more than oval in shape.

From the foregoing, the use of round wire is immediately suggested as having the advantages of a square wire and being, in addition, easy to manufacture. (It is interesting to note that a historical case of a torsional hairspring involved the use of round catgut.)

Since $K = \dfrac{\pi d^4 G}{32L}$

for $KL = 0.00368$,

$d = 0.00795"$ - the diameter of round wire to produce approximately 50 cycles per second at $L = 1.000"$.

Round wire, however, presents one serious disadvantage; that of holding during adjustment. The design of a clamp to accomplish this, such as some sort of pin vise, would undoubtedly be rather complicated. An alternate solution to allow use of the round wire, and one apparently used by U. S. Time Corporation, is to fix the length, adjusting the beat rate by changing the balance arm moment of inertia. This requires that the value of L be carefully chosen for a given diameter of wire, for the range of rate adjustment achievable through varying the moment of inertia (I) is, with normal balance configurations, more limited than that obtainable by means of the more common spring-length adjustment. Though I varies as the square of the "radius of gyration" of the balance, mechanical limitations usually do not permit a large percentage of total balance mass to be adjustable in moment radius.

Figure #14 is a "free pendulum plot of frequency (F) versus moment of inertia (I) for an assumed spring constant K = 0.00368. The range of adjustment possible in the original Frankford design was approximately 4.8×10^{-7} slug-in.^{2} to 5.8×10^{-7} slug-in.^{2}; about 20%. This would allow an adjustment of only about 4 cycles per second. The above range would need to be increased, perhaps to a 10-cycle range. It should be noted that tuning by means of an I adjustment would necessarily be a static trial-and-error process, as opposed to the dynamic tuning possibility of a spring adjustment.

Tests made with the above springs resulted in the selection of the .010" x .0057" wire as this is the lowest h/b ratio that can be adequately secured during adjustment. Using this wire, a length of .740" resulted, and this may be considered as the final design.

No change was made in the moment of inertia range of the balance system other than to ease manufacture. This is described in paragraph 3.3.4.2

3.3.2.12 Other Design Considerations

All the variables listed and discussed above are concerned with theoretical design: nothing has been said of allowable tolerances in the fabrication of parts. In general, this subject can be dismissed with the comment that good military timer practice should be followed in this respect, and the original Frankford drawings reflect this standard in most cases.

One area requiring particular attention, however, is the variation to be allowed in the relative interference between scape wheel teeth and pallet and cylinder surfaces. There are a number of dimensions involved and it is easy to build up an excessive tolerance. This must be avoided if predictable results in the finished escapement are to be expected.

An interesting point may be observed in relation to clearance between the scape wheel tooth and the cylinder diameters. Much has been said about the questionable effect of "bouncing" at the end of the impulse phase and the desirable objective of keeping the clearances small to reduce the "free fall". It is therefore worthy to note that if any clearance at all is provided (and of course, there must be some), the motion of the scape wheel in transferring from the cylinder O.D. to the cylinder I.D. is greater than the following I.D. to O.D. transfer. This is inherent in the cylinder escapement: for the Frankford design the two scape wheel movements comprising one cycle are:

$$1/2(I.D.+O.D.)=1/2(.510+.767)=23.6^\circ$$

$$45^\circ-1/2(I.D.+O.D.)= \quad 45^\circ-23.6^\circ=21.4^\circ$$

This, shown empirically on Figure #7 (Drawing #1582-D-17), is not considered unduly damaging to the cylinder escapement concept, even though it is an inherent design defect.

3.3.3 Operation Of The Cylinder Escapement

As is the case with most mechanisms, the cylinder escapement, if fabricated in exact accord with theory, will perform perfectly and predictably when assembled and run. This, of course, is impossible to achieve in practice, since the design must incorporate intentional deviations from the ideal, such as tolerances, and the actual parts generally add additional inaccuracies, such as non-uniform materials or finishes.

3.3.3.1 Adjustments

To counteract these errors, various devices are employed, primary of which is the provision for making small adjustments to the finished escapement. In the final redesign of the Frankford escapement, three adjustments have been incorporated.

1. An adjustable spring length to set the beat rate precisely. This was also provided for in the original design and should be retained in any future consideration of this, or in fact, of any escapement.
2. An adjustable inertia of the balance system, also used primarily to set the beat rate but also having (questionable) possibilities in poising of this system. This device consists of movable weights

at the extremities of the balance arm. In the Frankford design the angular position of the cylinder with respect to the balance arm, is such that the whole assembly is statically unbalanced to begin with; a change has been made here by rotating the cylinder so that a line joining the two pallet faces is parallel with the axis of the balance arm, resulting in a statically balanced assembly. With this change the balance adjusting weights are of value only in setting of the beat rate, and since this is already accomplished by the spring length adjustment, the weights should eventually be eliminated.

The weights have been retained in the REL revised design for only one purpose; they afford a finer setting of the beat rate than does the spring adjustment alone and for study purposes this has some value. For production, though, they should, without question, be deleted.

3. An adjustable rotational position of the cylinder (carrying the whole balance system) with respect to the scape wheel. As noted in paragraph 3.3.2.12 there are factors at work tending to vary the time between impacts, resulting in alternate long and short beats. These can be corrected by rotating the cylinder with respect to the scape wheel tooth and monitoring the resulting beat pattern through a watchmaster. This operation, known to watchmakers as putting the escapement "in beat", is accomplished here by allowing the spring housing to be rotated slightly within its mounting.

   In an advanced production design, this too could be eliminated although it would be at the cost of increased difficulty of sub assembly, and a somewhat less "perfect" mechanism.

3.3.3.2 Lubrication

A second method of overcoming imperfections is the use of lubricants. As was pointed out previously (paragraph 3.3.2.7), lubrication was found to be essential to provide reasonable beat rate stability under various torque conditions. This has been plotted on Figure #8.

In addition, lubrication plays an important part in providing long life for the escapement. One cylinder has now been run for about ten hours with no real deterioration apparent-this has not been possible with unlubricated parts.

The type of lubricant is important. Figure #15 shows the results of a thirty day test on a cylinder using a fine watch oil and Figure #16 shows a comparative thirty day test on the same cylinder using a solid film lubricant. The fugitive nature of oil, being fluid and sensitive to small changes in environment is evidenced by a period of "settling" at the outset and (not evident in thirty days), deteoriation with time.

The solid film, electrofilm, or in this case, "poxylube", provides the lubricating qualities of oil with a very high degree of stability and long shelf life. As a result of these tests, the treatment of the working portions of the cylinder with such a solid film lubricant is considered of maximum importance.

3.3.3.3 Balance "Flutter"

Finally, all escapements depend to a certain degree on the inherent self correction of the balance system to maintain a fixed rate. The small imperfections in action are counterbalanced by a change in the amplitude of the system, resulting in what appears to be a fluttering end point in the balance travel. This is natural and should not be considered undesirable.

It might be mentioned here that the "double beat rate" observed by Frankford Arsenal in their tests of this escapement was probably a combination of too great a center distance between scape wheel and pallet, coupled with a severe out of beat condition. Such an erratic operation has been duplicated on the thirty times size model.

3.3.4 Manufacture of the Cylinder Escapement

To obtain the best results from testing models of the modified escapement, and to facilitate handling during those tests, the components which comprise the escapement were designed as a package which could be easily mounted to a single power source. This system will be continued during Phase II with the result that all tests will be run using the same gear train, thus eliminating many potential sources of error in results.

In designing this package, and in fabricating the parts, a number of changes were made which do not necessarily alter the operation of the device, but which are intended to ease manufacture or assembly or to make adjustments simpler and more precise. The more important of these are listed below.

3.3.4.1 Cylinder: This part should be made in two parts, consisting of one pivot and the cylinder barrel as one piece, and the opposite pivot as the other. The second pivot is then pressed into the I.D. of the cylinder. (Three part construction may also be used, but is less desirable from an assembly standpoint.)

The advantages of this method of manufacture are:

1. The cylinder I.D. is drilled instead of end milled, resulting in better concentricity between

it and the O.D., and closer possible tolerances and better finish within the cylinder.

2. The length of cylinder between pivots may be better controlled through assembly than by machining, thus giving better control of cylinder endshake.

3.3.4.2 Balance Arm Or Lever: The method of forming the ends of this part to contain the screw weights is difficult and expensive. In an advanced production design, these weights need not be adjustable, which eases the fabrication immensely, but for the present it is felt better to machine this part using nuts instead of screws for adjustment of inertia.

3.3.4.3 Hole Sizes: Many of the pivot hole sizes reflect the European centimeter scale, and require special drills, reams and gages in the United States. These have been changed to correspond with tools readily available.

3.4 Summary of Changes

In redesigning the Frankford Torsional Hairspring escapement, a number of changes have been made, as described in the preceeding section. In summary these are:

1. The spacing of the scapewheel and the cylinder should be reduced from 0.1626" to 0.158", nominal, between centerlines, and the parts must be so toleranced that the variation will not exceed 0.156" to 0.159" (paragraph 3.3.2.8).
2. A slight radius should be given the cylinder where the receiving pallet meets the cylinder outer diameter. This is to reduce wear on the cylinder (paragraph 3.3.2.9).
3. The hairspring cross section dimensions should be increased from .003" x .010" to .006" x .010" to provide the desired 100 beats per second (50 cps) with a longer spring. This will make adjustment easier, and less sensitive, and the escapement less subject to variation (paragraph 3.3.2.11).
4. The angular position of the cylinder mounted to the balance arm should be such that the resulting assembly is in static balance (paragraph 3.3.3.1).

5. The balance weights should be eliminated eventually, although they were retained in this study (paragraph 3.3.3.1).
6. Provision for adjusting the angular location of the hairspring - cylinder - balance assembly with respect to the scape wheel should be considered (paragraph 3.3.3.1).
7. The working portion of the cylinder and pallet faces should be treated with a dry film lubricant (paragraph 3.3.2.6 and paragraph 3.3.4.1).
8. The cylinder should be made in two parts (paragraph 3.3.4.1).
9. Dimensions, particularly of pivot holes, should be changed from the present converted centimeter scale to correspond with tools and gages readily available in the United States (paragraph 3.3.4.3).

4.0 TEST RESULTS

Testing during Phase I has been limited to bench tests under ideal conditions. The test fixture used was a precision 80:1 ratio gear train to which the cylinder escapement could be attached as a package assembly. Power was applied to the escapement by a falling weight drive on the gear train, providing constant torque at all times.

A unit fabricated to the final design, with the design modifications listed in paragraph 3.4 was extensively tested using this arrangement, monitoring each test on the recording watchmaster. The total running time on this unit is in excess of ten hours, with no appreciable wear on the cylinder or scape wheel.

The significant test results are recorded and plotted in this report, and some discussion of these results appears in the preceeding sections.

4.1 Torque Tests

Results of tests run at various torques on both a dry cylinder and the same cylinder treated with a dry film lubricant are listed on Table #2 and plotted on Figure #8. During this test it was seen that the escapement, while quite torque sensitive when run dry, is capable of the ±0.2% tolerance required within the 15 to 25 in-oz. range when lubricated. This has been discussed in paragraph 3.3.2.7.

4.2 Duration Tests

A long series of tests were made in which the escapement was adjusted and run over a period of about thirty days without changing the adjustment. This was first done with the cylinder oiled with a fine watch oil, and then repeated with the same escapement in which the cylinder was first treated with a dry film lubricant. Results of the first of these series are given on Table #3 and plotted on Figure #15. The second series is recorded on Table #4 and plotted on Figure #16. It can be seen that the oiled cylinder operated fairly well over the last 25 days of the tests (within .23% total tolerance) but required an initial period for the oil to settle during which the escapement behaved somewhat erratically. This parallels past experience with timers using oil as a lubrication. In addition, the disadvantageous effects of shelf life on oil is well known, so that it is generally avoided in military devices.

With the cylinder treated with the dry film lubricant, on the other hand, the escapement operated within a total tolerance of about 0.17% throughout the 27 days of operation. On the basis of this test, the decision to recommend this type of lubrication as a permanent change was made.

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TABLE I

Comparison of Escapement Types

Note: Dimensions are nominal or measured

ESCAPEMENT	CYLINDER	CYLINDER	DETACHED	JUNGHANS	REF. ¶
Source	Frankford Arsenal (Tavero)	France C. 1830	R.E.L. (Hamilton)	R.E.L.	
Ref. Figure	4 & 5	2	3 & 5	5	
Scape Wheel					
Diameter	.3307"	.3370"	.3440"	.3260"	3.3.2.1
No. of Teeth	8	15	15	20	3.3.2.2
Tooth Pitch	45°	24°	24°	18°	3.3.2.2
Rate-Cycles/Sec.	50	2.5	30	98.8	3.3.2.2
Rate-Beats/Sec.	100	5	60	197.6	3.3.2.2
Impacts/Tooth/Min.	750	20	240	593	3.3.2.2
Impacts/Tooth/Min. @ 100 Beats/Sec.	750	400	400	300	3.3.2.2
ARC Length/Tooth	.1298"	.0705"	N.A.	N.A.	3.3.2.3
Tooth Length	.0476"	.0255"	N.A.	N.A.	3.3.2.3
Tooth % of ARC	36.7%	36.2%	N.A.	N.A.	3.3.2.3
Space between Teeth	.0822"	.0450"	N.A.	N.A.	3.3.2.3
Cylinder O.D.	.0767"	.040"	N.A.	N.A.	3.3.2.3
Cylinder I.D.	.0510"	.030"	N.A.	N.A.	3.3.2.3
Cylinder Wall	.0129"	.005"	N.A.	N.A.	3.3.2.3
Clearance - I.D.	.0034"	.0045"	N.A.	N.A.	3.3.2.3
Clearance - O.D.	.0055"	.0050"	N.A.	N.A.	3.3.2.3
Scape Wheel inertia ($SW6-IN^2$)	7.68×10^{-8}	---	6.62×10^{-8}	7.33×10^{-8}	3.3.2.5
Scape Wheel material	Stain.Steel	Hard.Steel	Lead.Brass	Hard.Steel	3.3.2.6
Pallet Material	Stain.Steel	Hard.Steel	Jewels	Stain.Steel	3.3.2.10
Balance Inertia ($Slug-In^2$)	4.8 to 5.8 $\times 10^{-7}$	1.3×10^{-6}	5.4×10^{-7}	6.3×10^{-7}	3.3.2.11

TABLE II

Torque vs Beat Rate

No	Date 1961	gms	BPS	Note	No	Date 1961	gms	BPS	Note
1	5-5	150	97.8	Dry Cylinder	4	7-12	200	98.650	Dry Film Lube
2	5-5	200	98.3	" "	5	7-12	250	98.580	"
3	5-5	250	98.5 - 98.7	" "	6	7-12	300	98.620	"
4	5-5	300	98.772	" "	7	7-12	350	98.800	"
5	5-5	400	99.2	" "	8	7-12	400	98.848	"
6	5-5	450	99.5	" "	9	7-12	450	99.304	"
7	5-5	500	102.5	" "	10	7-12	500	99.532	"
8	5-8	100	96.4	" "	11	7-12	550	99.684	"
9	5-8	150	97.2	" "	12	7-12	600	99.836	"
10	5-8	200	97.8	" "	13	7-12	550	99.532	"
11	5-8	250	98.1	" "	14	7-12	500	99.304	"
12	5-8	300	98.772	" "	15	7-12	450	99.076	"
13	5-8	350	98.9	" "	16	7-12	400	98.848	"
14	5-8	400	99.2	" "	17	7-12	350	98.696	"
15	5-8	450	101.3	" "	18	7-12	300	98.468	"
16	5-8	500	102,6	" "	19	7-12	250	98.350	"
17	5-8	550	103.7	" "	20	7-12	200	98.240	"
18	5-8	600	105.7	" "	21	7-12	150	98.392	"
					22	7-12	100	98.200	"
1	7-12	50	97.0	Dry Film Lube	23	7-12	50	97.100	"
2	7-12	100	98.544	"					
3	7-12	150	98.924	"					

TABLE III

Duration Test 6-7-61 to 7-7-61

(Oiled Cylinder)

No.	Date 1961	Time	B.P.S.	Note
1	6-7	--	98.772	Dry
2	6-8	9:15	98.848	Oil
3	6-8	11:30	98.924	"
4	6-8	1:00	98.848	"
5	6-8	2:15	98.696	"
6	6-8	2:45	98.772	"
7	6-8	4:00	98.620	"
8	6-9	8:00	98.848	"
9	6-9	9:30	98.924	"
10	6-9	12:00	98.924	"
11	6-9	2:00	98.924	"
12	6-9	3:30	98.848	"
13	6-12	8:30	99.228	"
14	6-12	10:00	99.304	"
15	6-12	12:00	99.152	"
16	6-12	2:00	99.076	"
17	6-12	3:45	99.152	"
18	6-13	8:00	99.304	"
19	6-13	1:00	99.228	"
20	6-13	3:30	99.228	"
21	6-14	8:00	99.228	"
22	6-15	1:00	99.304	"
23	6-19	8:00	99.304	"
24	6-22	3:00	99.228	"
25	7-7	10:00	99.266	"

TABLE IV

Duration Tests 7-14-61 to 8-10-61

(Dry Film Lubricant on Cylinder)

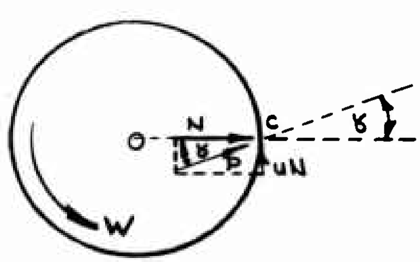
No.	Date	Time	B.P.S.
1	7-14	2:00	98.738
2	7-14	2:15	98.851
3	7-14	3:00	98.845
4	7-17	9:45	98.810
5	7-17	11:00	98.804
6	7-17	1:30	98.908
7	7-18	8:45	98.775
8	7-18	11:15	98.830
9	7-18	1:15	98.844
10	7-18	3:45	98.794
11	7-19	9:00	98.783
12	7-19	1:15	98.833
13	7-19	3:45	98.829
14	7-20	8:45	98.810
15	7-20	11:30	98.806
16	7-20	3:00	98.764
17	7-21	9:15	98.764
18	7-21	11:30	98.843
19	7-21	2:00	98.830
20	7-21	3:30	98.814
21	7-31	8:30	98.772
22	7-31	2:00	98.848
23	8-1	9:30	98.830
24	8-1	3:30	98.833
25	8-2	8:30	98.810
26	8-2	3:30	98.764
27	8-3	9:00	98.775
28	8-3	3:30	98.833
29	8-4	9:00	98.833
30	8-4	3:30	98.806
31	8-7	8:30	98.845
32	8-10	2:00	98.772

APPENDEX A

Friction Drag on Cylinder During Dead-Beat Phase

It has been brought out that our escapement timing rate may be caused to vary solely through fluctuation or drift of those external disturbances which cause the control balance to deviate from an ideal "free pendulum balance". Three discreet disturbances were recognized: (1) Cylinder bearing friction; (2) Impulse pattern; (3) Dead-beat drag. Optimization of timing rate suggested two possible objectives: (1) Minimization of deviations or disturbances; (2) Balancing of parameter fluctuation tendencies so as to cancel out possible net change effects.

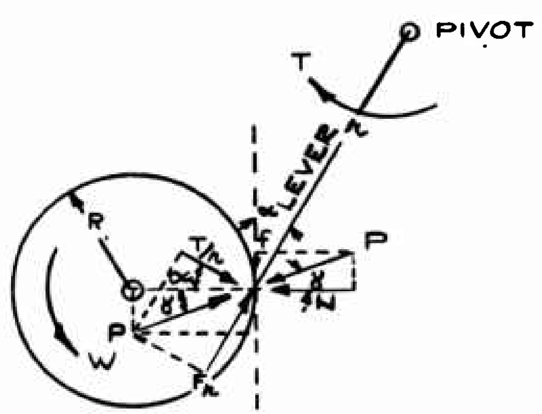
The following discussion is a general examination of friction loading during the head-beat or "rest" phases, during which a wheel tooth rests "dead" on a cylindrical face of the moving balance system.



μ = friction coefficient between contacting surfaces.

Let $\tan \gamma = \mu$

Any force pressing against a rotating cylinder at a contact point (c), by Newton's 2nd Law, must be opposed by an equal and opposite "reaction force". This reaction force can be thought of as consisting of a radial and a tangential component, relative to the cylinder. But the only tangential component which may arise - for a contact point (c) which is fixed in space - is the frictional force (μN), where N is the normal (or in this case, radial) component. It can be seen, therefore, that the resultant reaction force (P) must lie along a line whose direction relative to the normal (N) is determined by the friction angle (γ), regardless of absolute magnitude or apparent contact angle. Since reaction equals action, this means that a non-moving external force can only be applied in the unique direction corresponding to the friction angle. Otherwise, the applied force could not maintain a fixed position in space! (Of course, if cylinder were not moving, the static friction coefficient and equivalent friction angle would not be unique.)



If, for example, a pivoted lever of length (r) is caused to press against point (c) by means of a torque (T), as shown, the cylindrical surface must provide a reaction component (T/r) perpendicular to the direction of r, in order to hold the lever in a rest position.

But, $T/r = P\cos(\alpha+\gamma)$, where α is the angle the lever makes with the tangent to the cylinder at the contact point, as shown.

Thus, $P = \dfrac{T}{r\cos(\alpha+\gamma)}$

And, $N = P\cos\gamma = \dfrac{T\cos\gamma}{r\cos(\alpha+\gamma)}$ and $\therefore$ Friction $(f) = \mu N = N\tan\gamma = P\sin\gamma$

Thus, friction torque on cylinder $(T_f) = RN\tan\gamma = \dfrac{RT\sin\gamma}{r\cos(\alpha+\gamma)}$

Also, the lever is subjected to a radial force $(F_r) = P\sin(\alpha+\gamma) =$

$T/r\,\tan(\alpha+\gamma)\,\}$ Compression if $(\alpha+\gamma)$ is positive

It is seen that as $(\alpha+\gamma)$ approaches $\pi/2$, friction torque (T_f) and total bearing loads (P) become extremely high. (This condition is equivalent to allowing the lever to assume the direction of the friction angle.) In effect, instantaneous "binding" or "locking" results.

If $\alpha = -\gamma$, the friction angle takes a direction perpendicular to the lever, and $P = T/r$, $N = T/r\cos\gamma$, $T_f = -\dfrac{RT\sin\gamma}{r}$, $F_r = 0$.

This direction would <u>appear</u> to offer <u>minimum</u> friction torque and bearing loads. However, if ω should then reverse its direction, γ is effectively reversed in sign, and <u>loading increases</u> to that of (1), above, with the exception that f, T_f, and F_r are reversed in direction.

If $\alpha = 0$, lever is tangent to cylinder, and $P = \dfrac{T}{r\cos\gamma}$, $N = T/r$,

$T_f = \dfrac{RT}{r}\tan\gamma$, $F_r = T/r\tan\gamma = T_f/R$.

The interesting thing about this case is that loading remains unchanged in magnitude (not direction) as ω reverses sing. It is the only lever position which produces this net effect. Any other lever angle would offer less resistance and bearing load for one direction of ω than for the reverse ω. For the direction of ω whown above, forces could be reduced to a minimum by letting $\alpha = -\gamma$ but, as discussed, upon reversal of ω loading would become worse than for the tangential lever position. Thus, with an oscillating cylinder, the tangential position would appear to offer an optimum compromise.

The analysis above can be applied to the action of a wheel tooth on either the outside or inside cylindrical surface of the Frankford escapement. If, to begin with, we accept the Frankford escapewheel configuration, it can be shown that the original cylinder escapewheel separation distance (0.1626 inches) was too great. In the first place, the REL 30 X Model showed operation to be marginal or "run-a-way" with this separation. An optimum separation can be determined on the basis of minimum frictional drag. The tangential position previously referred to as a best choice corresponds to the requirement that the line of action of each wheel tooth, in the "dead" position, pass through the pallet pivot. Unfortunately, with a finite cylinder thickness, it is impossible to satisfy this requirement simultaneously for both the inside and outside cylinder diameters. (An infinitely thin cylinder wall would be necessary.) Figure 9 demonstrates that a separation of about 0.158 inches would compromise this dilemma most effectively.

It should be noted that the foregoing analysis is quite independent of the extent to which the cylindrical wall is "cut-away" and equally independent of pallet shape. In contrast, the basic cylinder diameters are of direct importance. However, other considerations fairly well fix these diameters. For a given escapewheel, the minimum I.D. and maximum O.D. are determined as "ideal" limitations. Production tolerance limitations make it necessary to open-up this ideal I.D. and reduce the O.D. In order to hold "drop" impact to a minimum, it is desirable to deviate as little as possible from the ideal. When the original Frankford cylinder is moved in to a separation distance of 0.158 inches, there would appear to be little room for improvement through readjustment of cylinder diameters, in view of reasonably achievable production tolerances. Moreover, the preceding analysis, though critical with respect to small variations in separation distance, is relatively insensitive to slight changes in chlinder diameter values.

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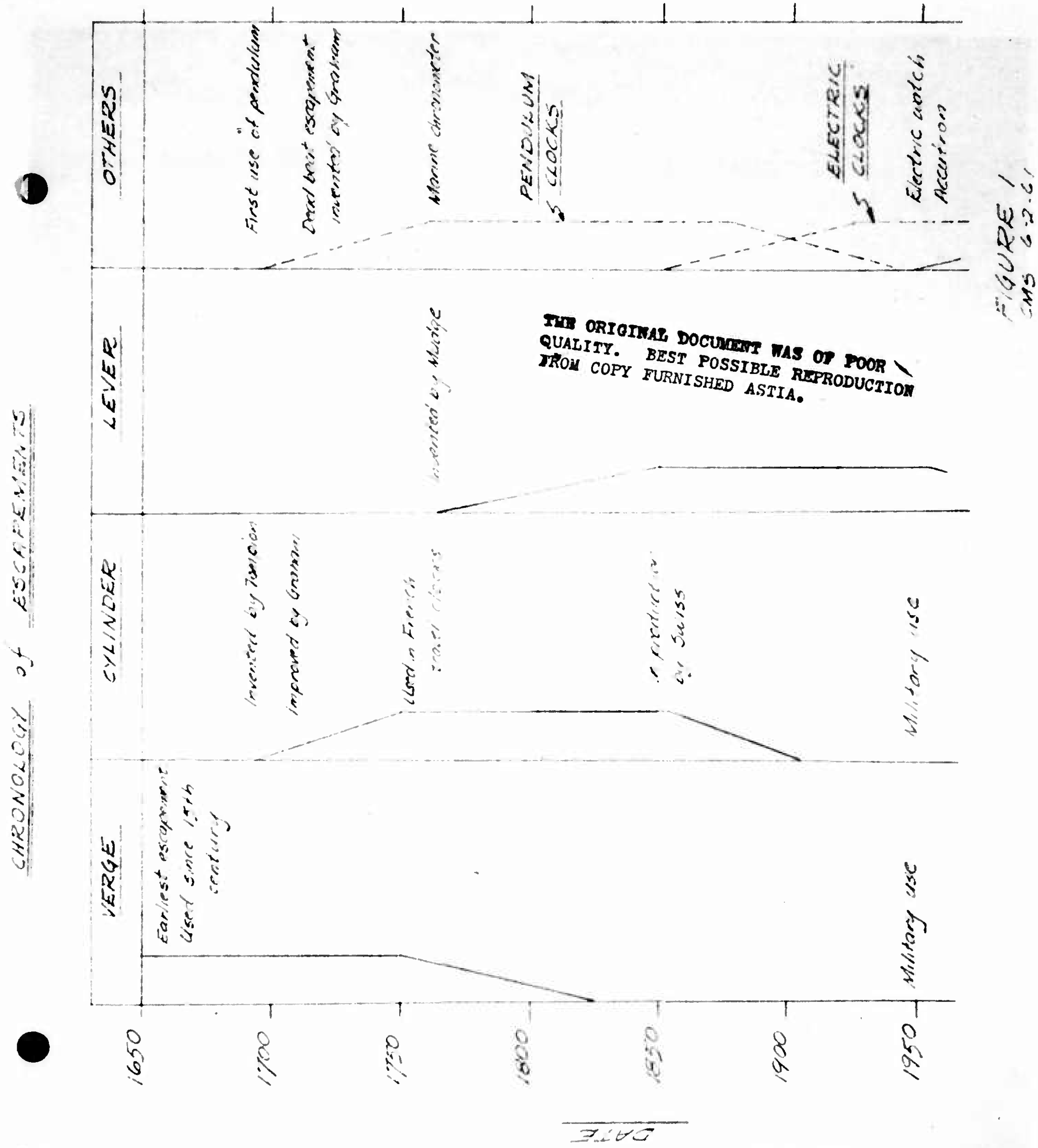
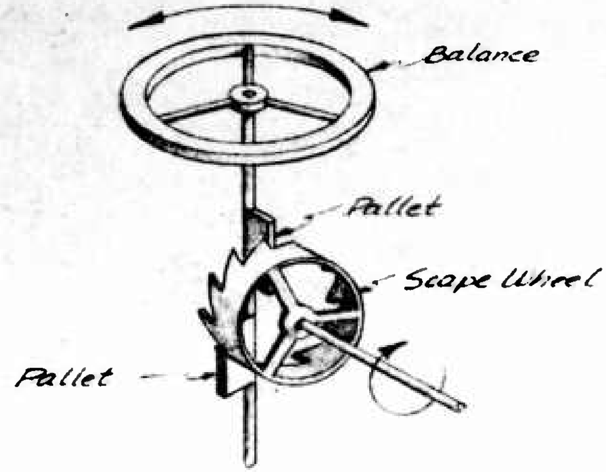
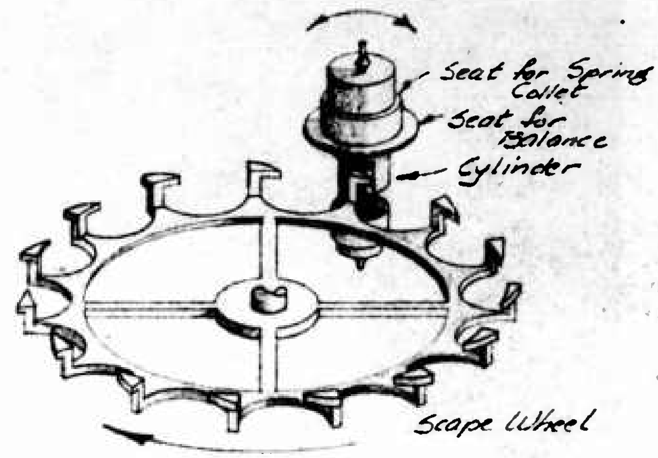


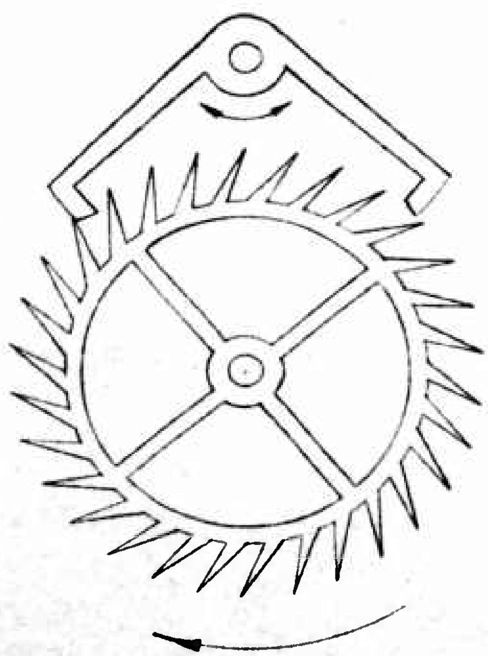
FIGURE 1
ESCAPEMENT CHRONOLOGY



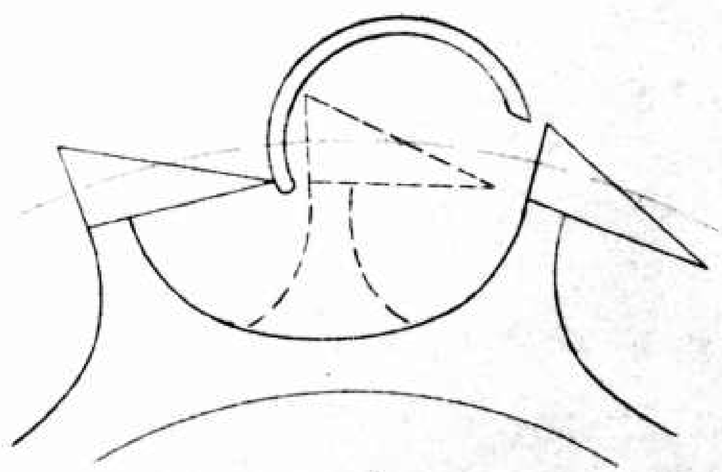
EARLY VERGE ESCAPEMENT



CYLINDER ESCAPEMENT



GRAHAM DEAD BEAT ESCAPEMENT



CYLINDER ESCAPEMENT DETAIL

FIGURE 2
HISTORICAL ESCAPEMENTS

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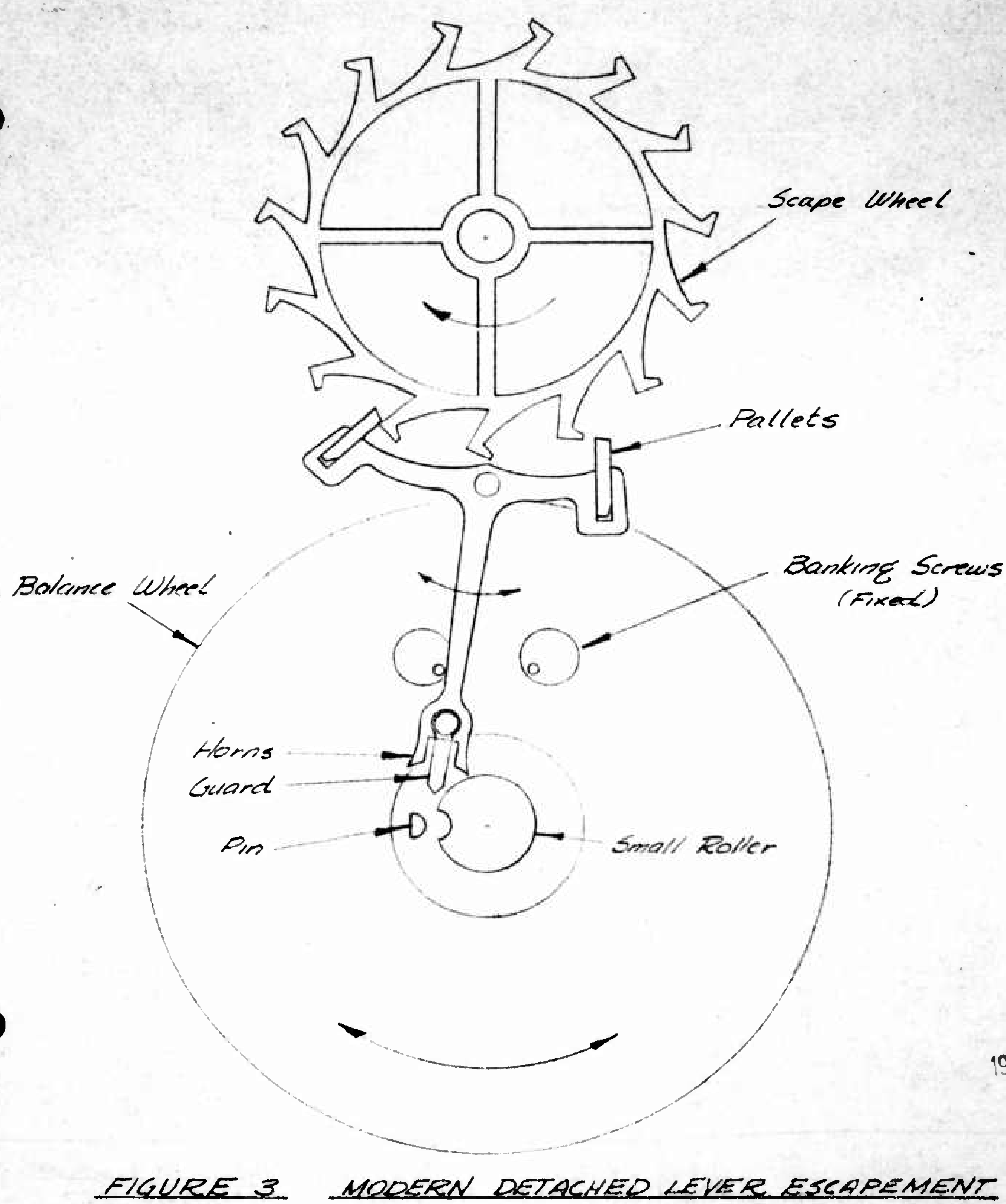
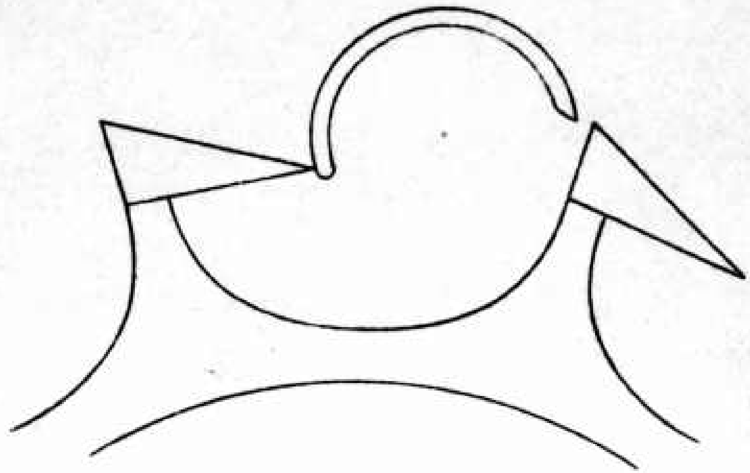


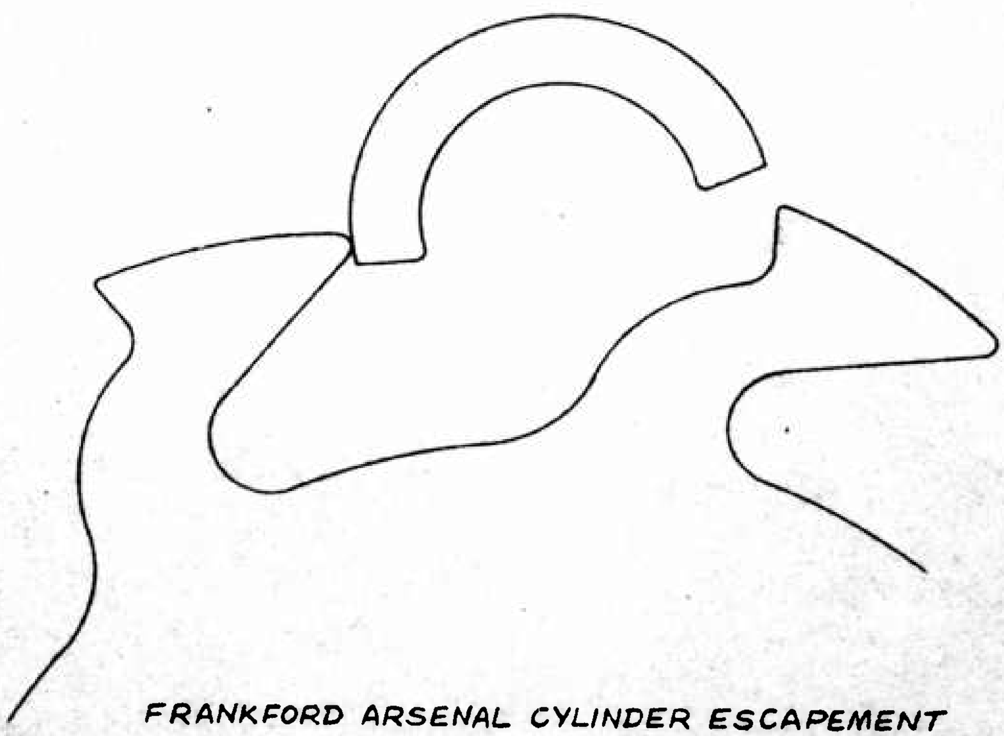
1961

FIGURE 3 MODERN DETACHED LEVER ESCAPEMENT

CMS 6-5-61



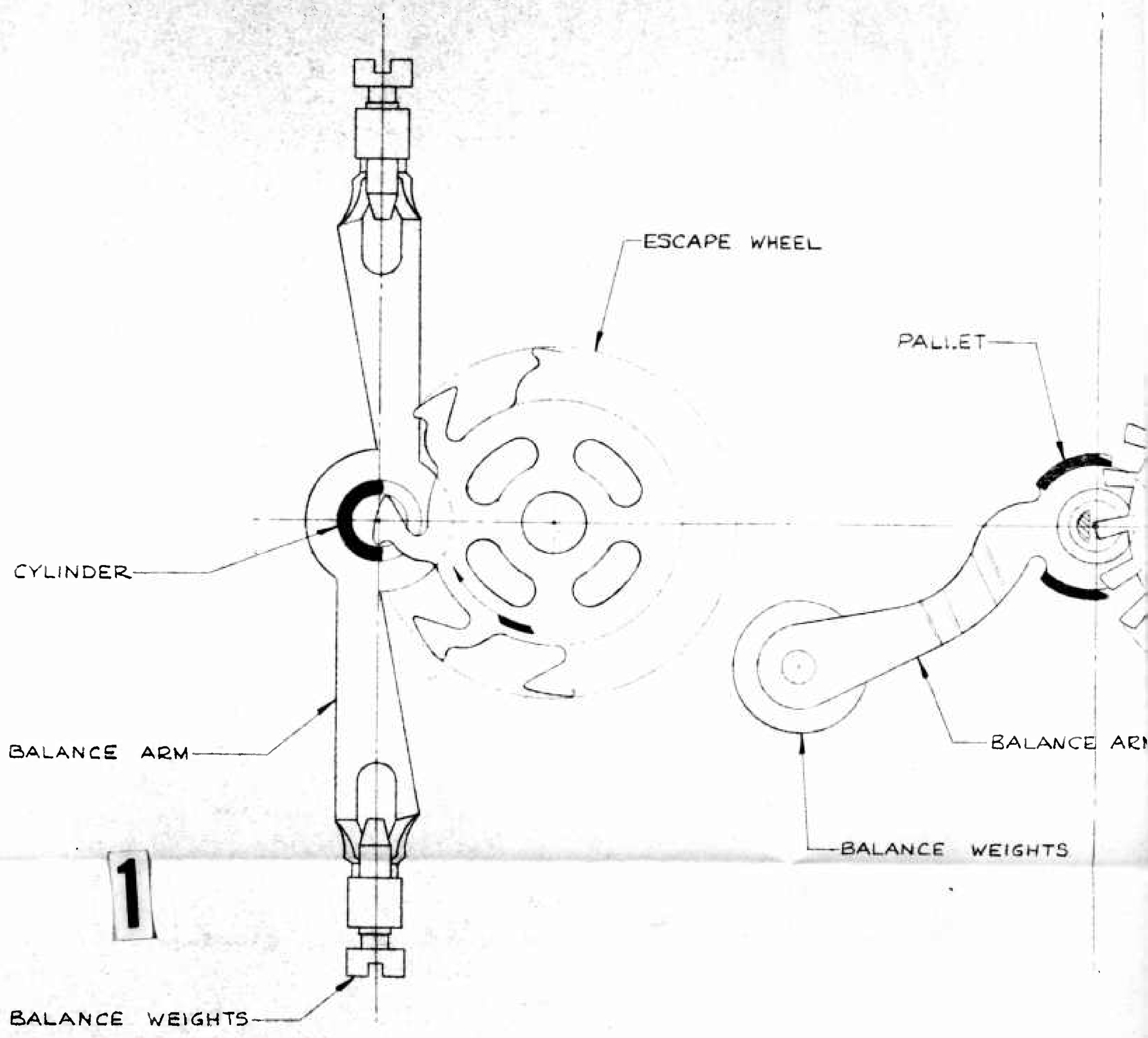
HISTORICAL CYLINDER ESCAPEMENT



FRANKFORD ARSENAL CYLINDER ESCAPEMENT

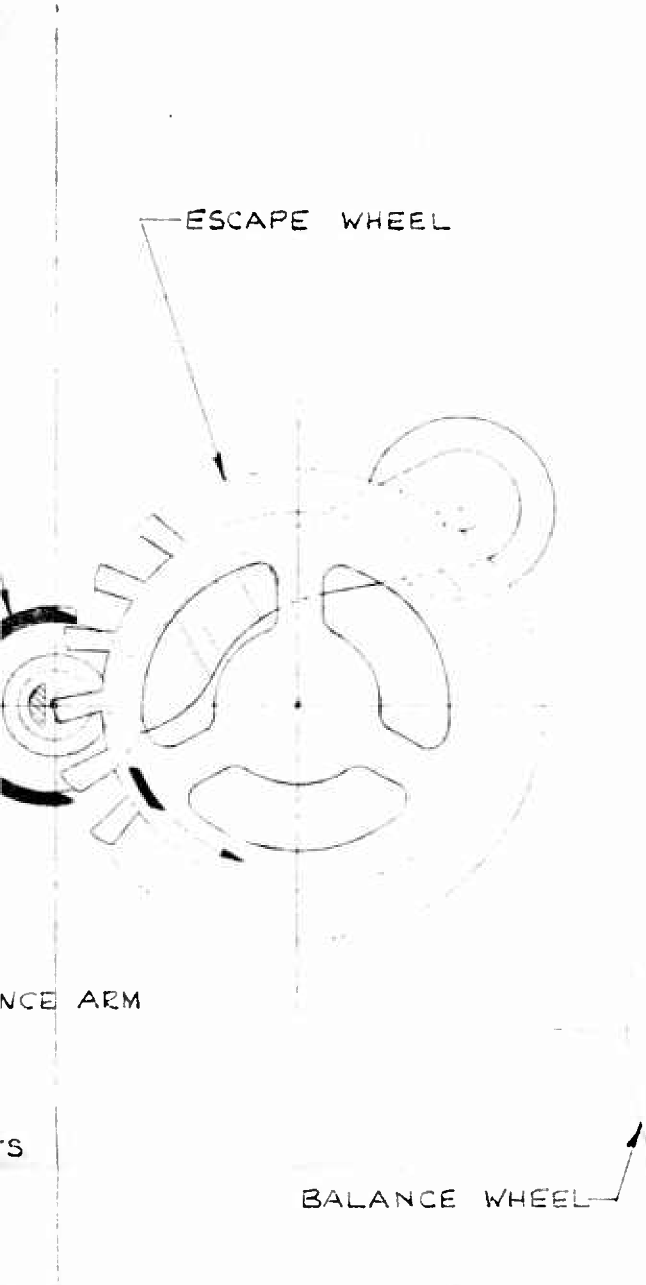
FIGURE 4
COMPARISON OF CYLINDER ESCAPEMENTS

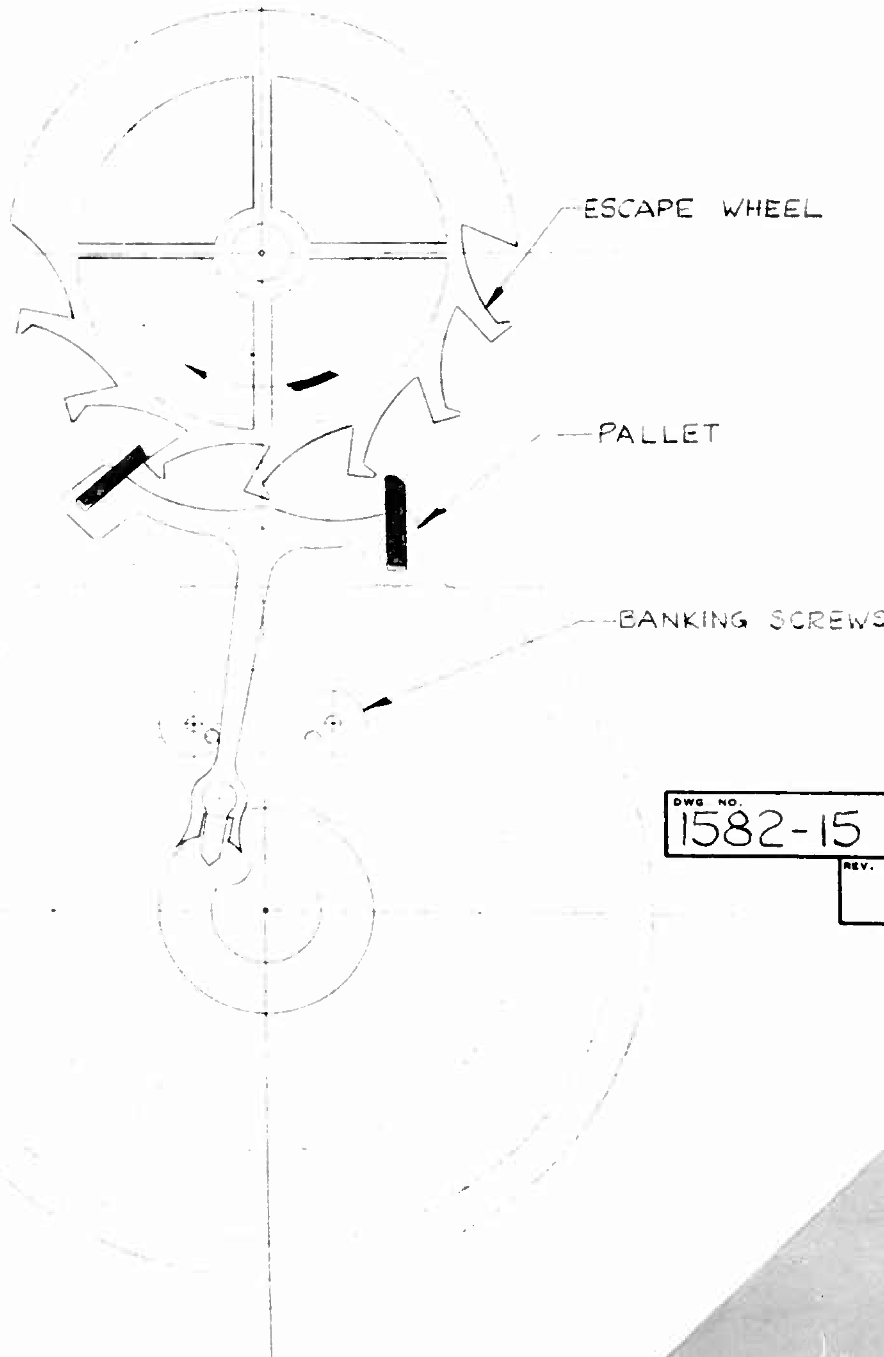
 R.H.P. 7-19-61



TAVERO ESCAPEMENT

REVISIONS		DWG. NO. 1582-15	REV.
SYM.	DESCRIPTION	BY	DATE

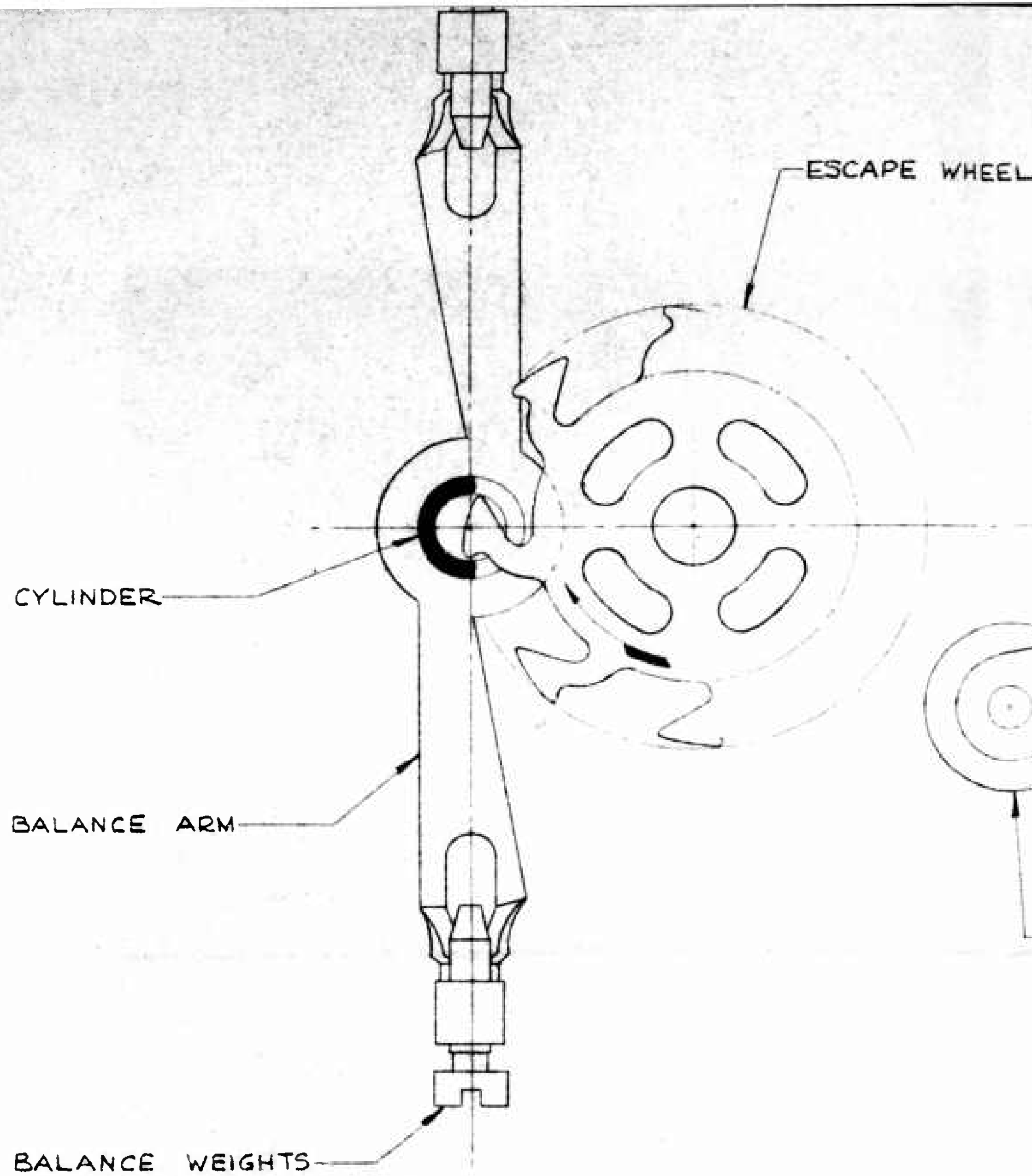




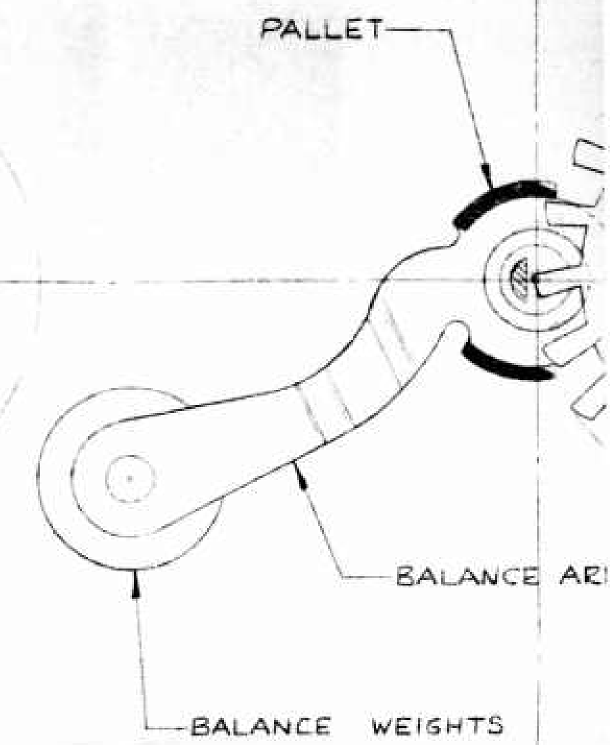
DWG. NO. 1582-15

REV.

2

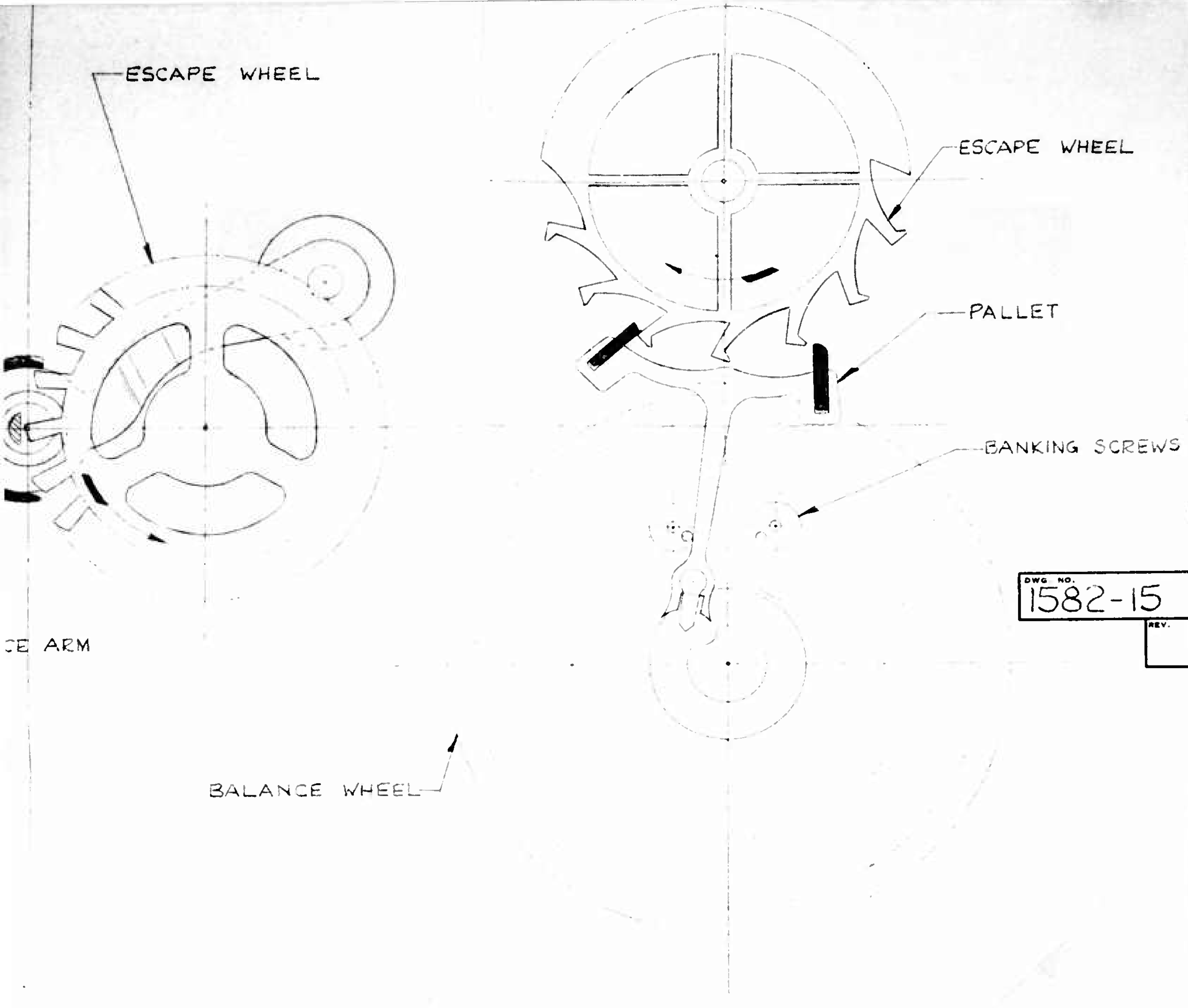


TAVERO ESCAPEMENT



R.E.L. JUNGHANS E

3



DWG. NO. 1582-15

REV.

NS ESCAPEMENT

R.E.L. DETACHED LEVER ESCAPEMENT

4

FIGURE 5

LIMITS OF ACCEPTABLE WORKMANSHIP ARE DEFINED IN MP-2000

TOLERANCES (UNLESS OTHERWISE SPECIFIED)	NEXT ASS'Y.	NO. REQ'D.	RAYMOND ENGINEERING LABORATORY, INC. MIDDLETOWN, CONNECTICUT		W. O.	TITLE ESCAPEMENT COMPARISON			
DECIMAL DIM. ± .005					NO. REQ'D.				
FRACTIONAL DIM. ± 1/64			MATERIAL	FINISH	DRAWN GENE. C	DATE 5-3-61	SCALE 10:1		
ANGULAR DIM. ± 5°			SPEC.	SPEC.	CHECKED	DATE	DWG. SIZE C	DWG. NO. 1582-15	REV.
ALL DIMENSIONS GIVEN IN INCHES					APPROVED	DATE			

PRINT ISSUED AUG 10 '61

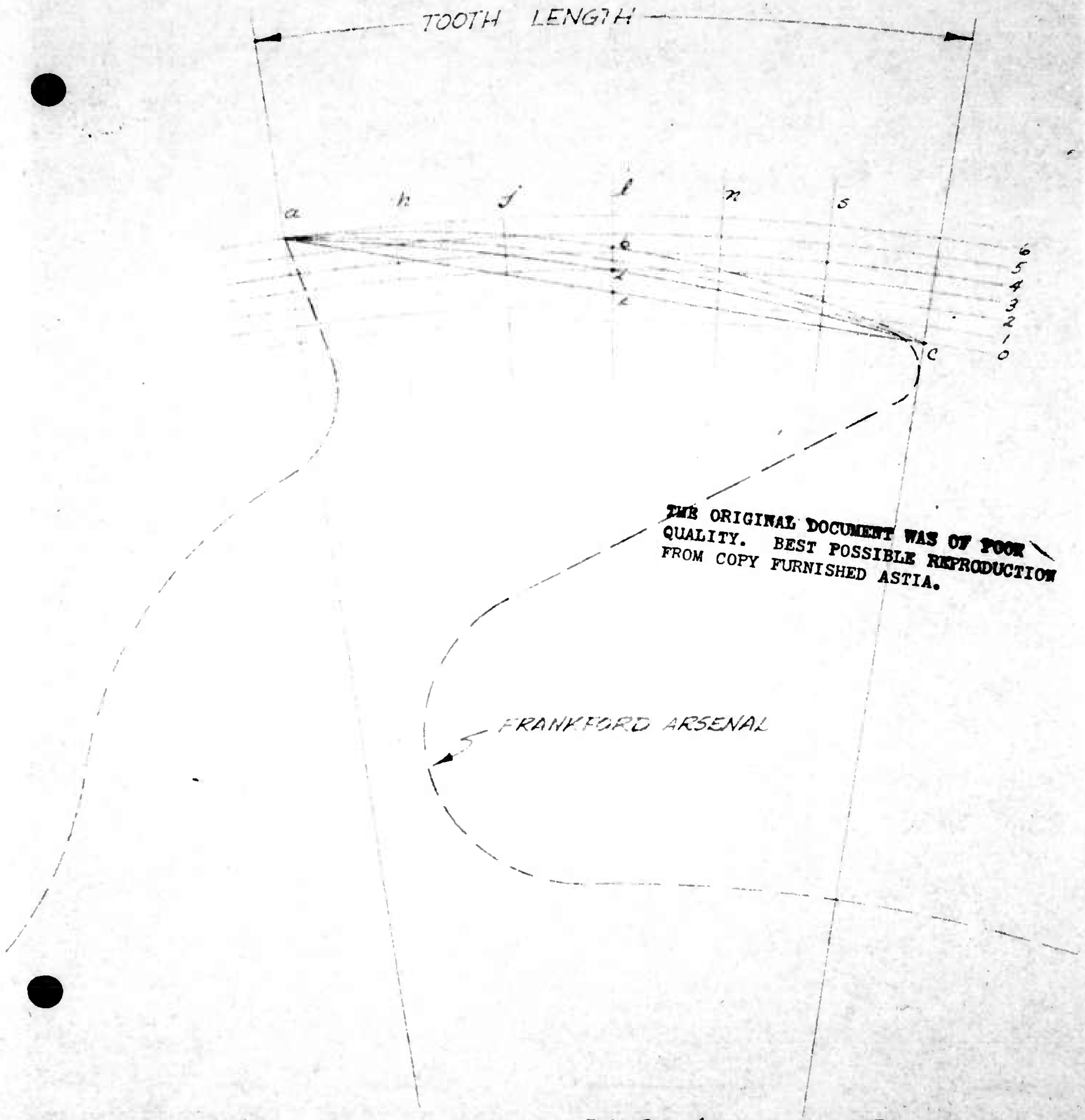


FIGURE 6 50:1
SCAPE WHEEL IMPULSE GEOMETRY
CMS 7-20-61

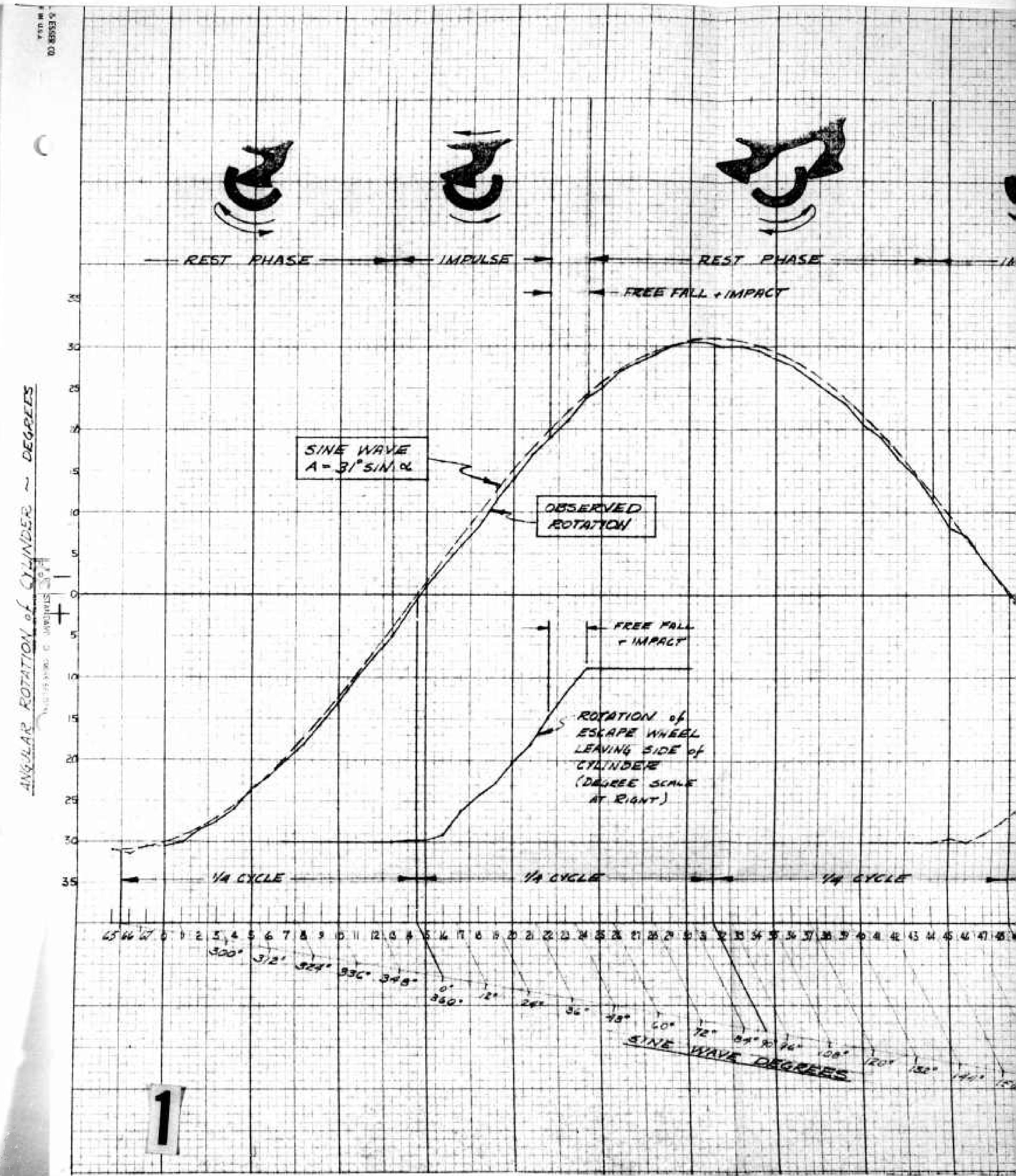

ANGULAR ROTATION of CYLINDER ~ DEGREES
REST PHASE
IMPULSE
REST PHASE
FREE FALL + IMPACT
SINE WAVE
A = 31° SIN. α
OBSERVED ROTATION
FREE FALL + IMPACT
ROTATION of ESCAPE WHEEL LEAVING SIDE of CYLINDER (DEGREE SCALE AT RIGHT)
1/4 CYCLE
1/4 CYCLE
1/4 CYCLE
300°
312°
324°
336°
348°
0°
360°
12°
24°
36°
48°
60°
72°
84°
90°
96°
108°
120°
132°
SINE WAVE DEGREES
1

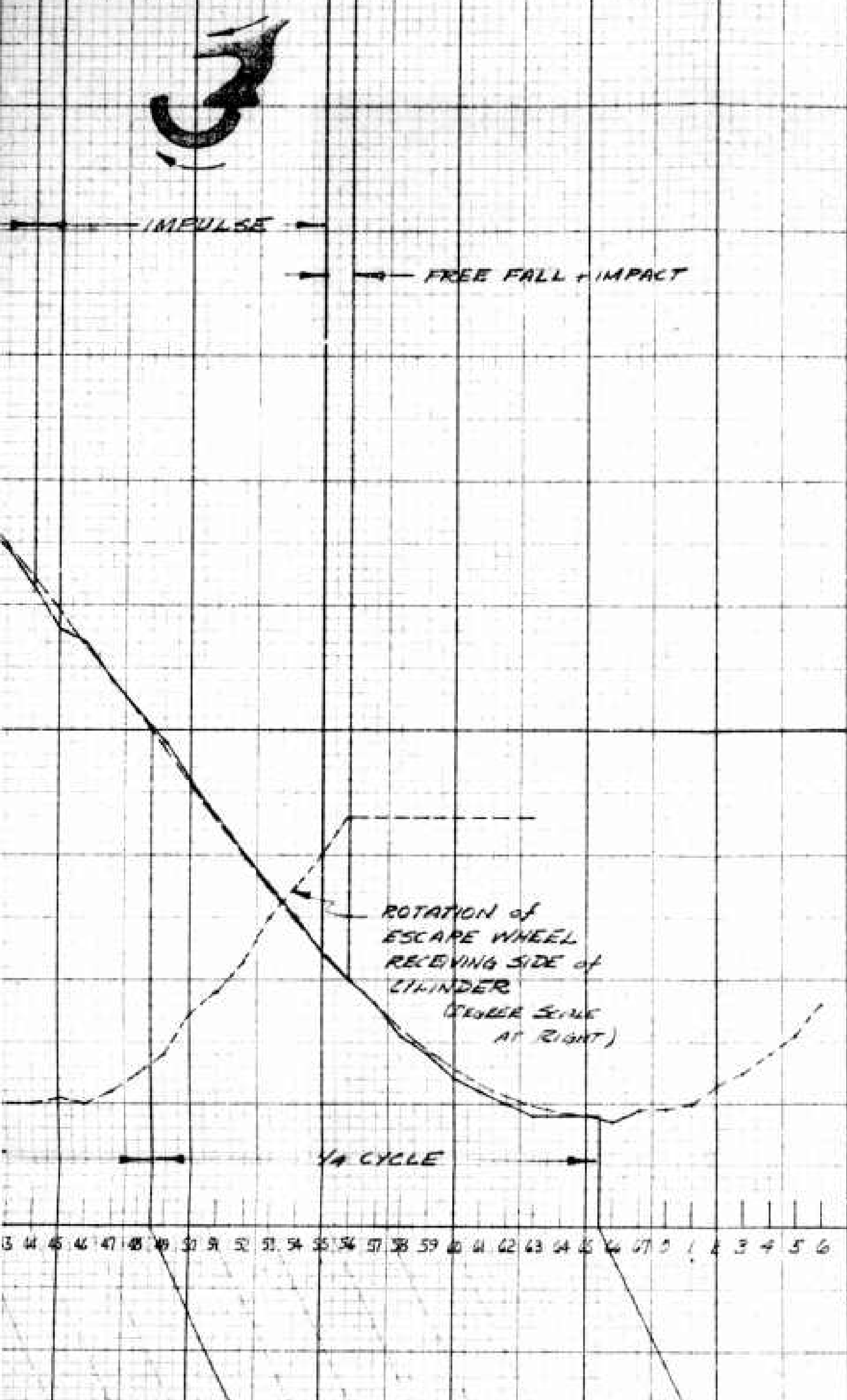
FIGURE 7 1582-D-17

TAVERO ESCAPEMENT

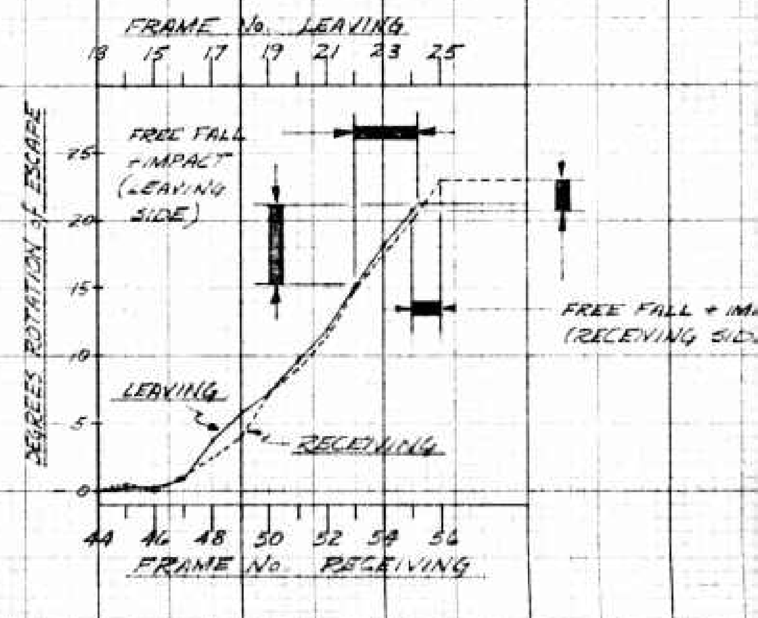
CYLINDER ROTATION vs TIME

Based on 30x MODEL TESTS

CMS 18 MAY 1961



COMPARISON of ESCAPE WHEEL ROTATION DURING IMPULSE and FREE FALL PHASES



2

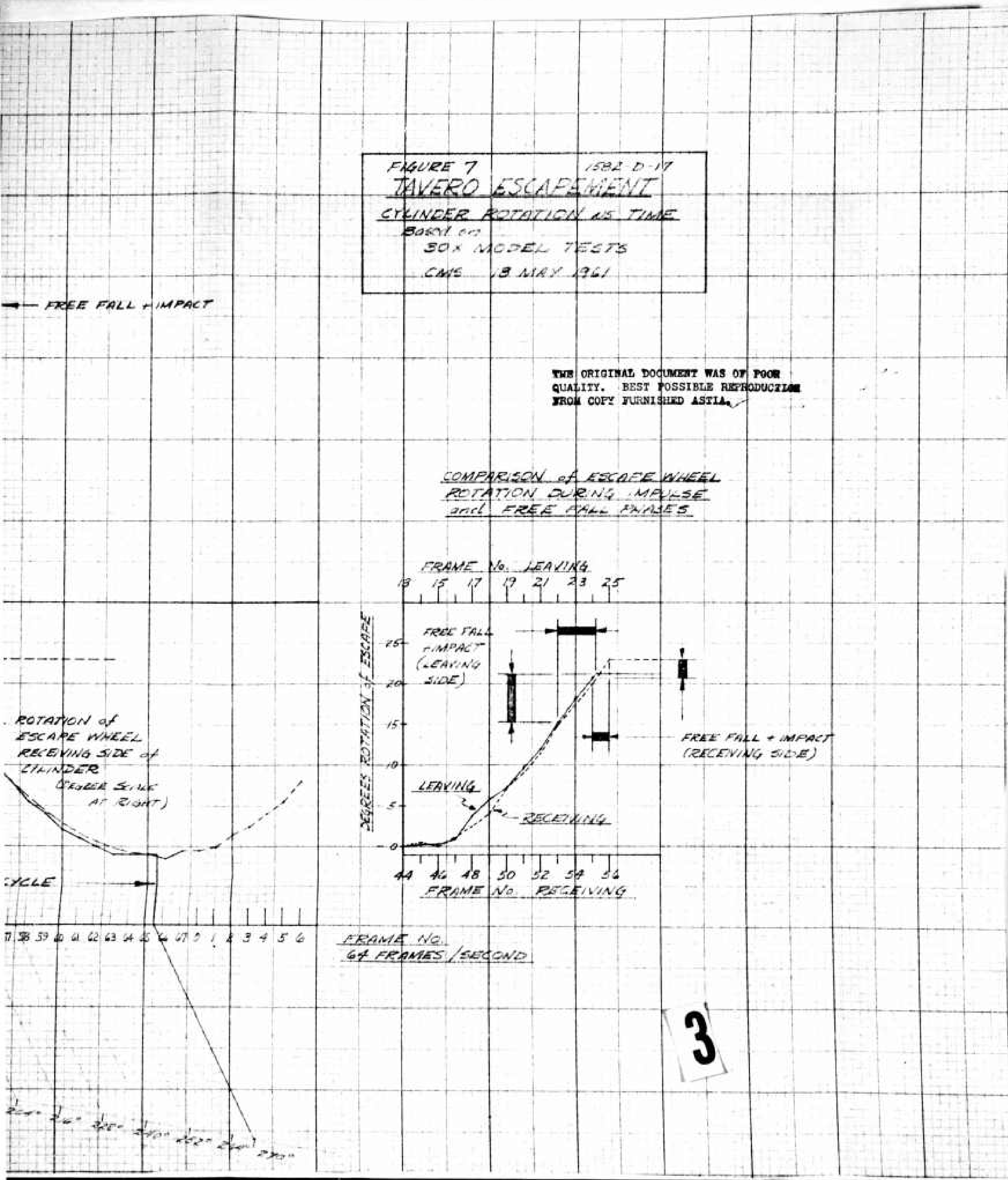
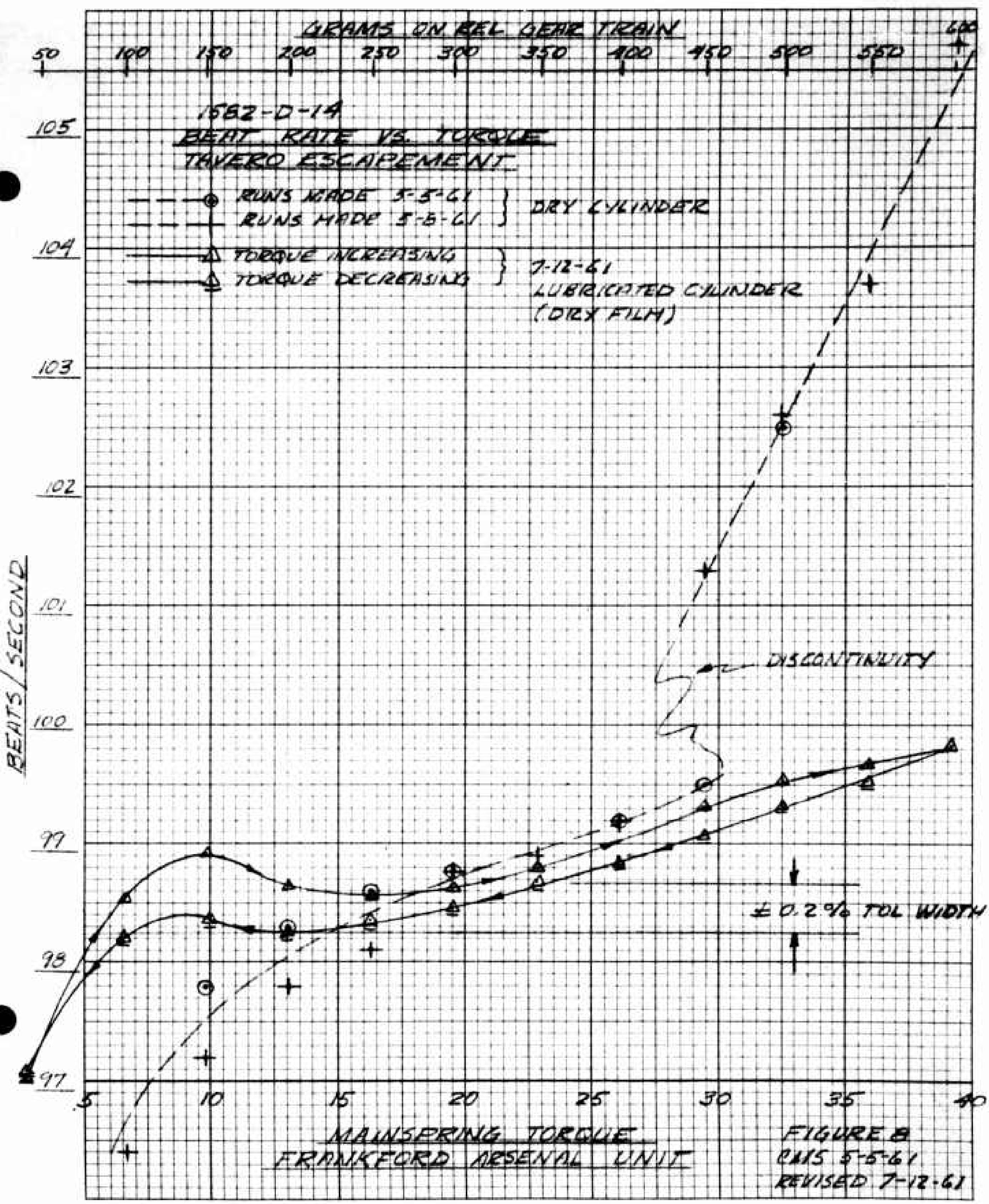


FIGURE 7 1582-D-17
JAVERO ESCAPEMENT
CYLINDER ROTATION vs TIME
Based on
30x MODEL TESTS
CMS 18 MAY 1961
FREE FALL + IMPACT
COMPARISON of ESCAPE WHEEL ROTATION DURING IMPULSE and FREE FALL PHASES
FRAME No. LEAVING
13 15 17 19 21 23 25
DEGREES ROTATION of ESCAPE
25 20 15 10 5 0
FREE FALL + IMPACT (LEAVING SIDE)
LEAVING
RECEIVING
FREE FALL + IMPACT (RECEIVING SIDE)
44 46 48 50 52 54 56
FRAME No. RECEIVING
ROTATION of ESCAPE WHEEL RECEIVING SIDE of CYLINDER (DEGREE SCALE AT RIGHT)
CYCLE
57 58 59 60 61 62 63 64 65 66 67 0 1 2 3 4 5 6
FRAME No.
64 FRAMES/SECOND
3



GRAMS ON REL GEAR TRAIN
50
100
150
200
250
300
350
400
450
500
550
600
1582-D-14
BEAT RATE VS. TORQUE
TAVERO ESCAPEMENT
RUNS MADE 5-5-61
RUNS MADE 5-5-61
DRY CYLINDER
TORQUE INCREASING
TORQUE DECREASING
7-12-61
LUBRICATED CYLINDER
(DRY FILM)
105
104
103
102
101
100
99
98
97
BEATS / SECOND
DISCONTINUITY
± 0.2% TOL WIDTH
5
10
15
20
25
30
35
40
MAINSPRING TORQUE
FRANKFORD ARSENAL UNIT
FIGURE 8
CJIS 5-5-61
REVISED 7-12-61

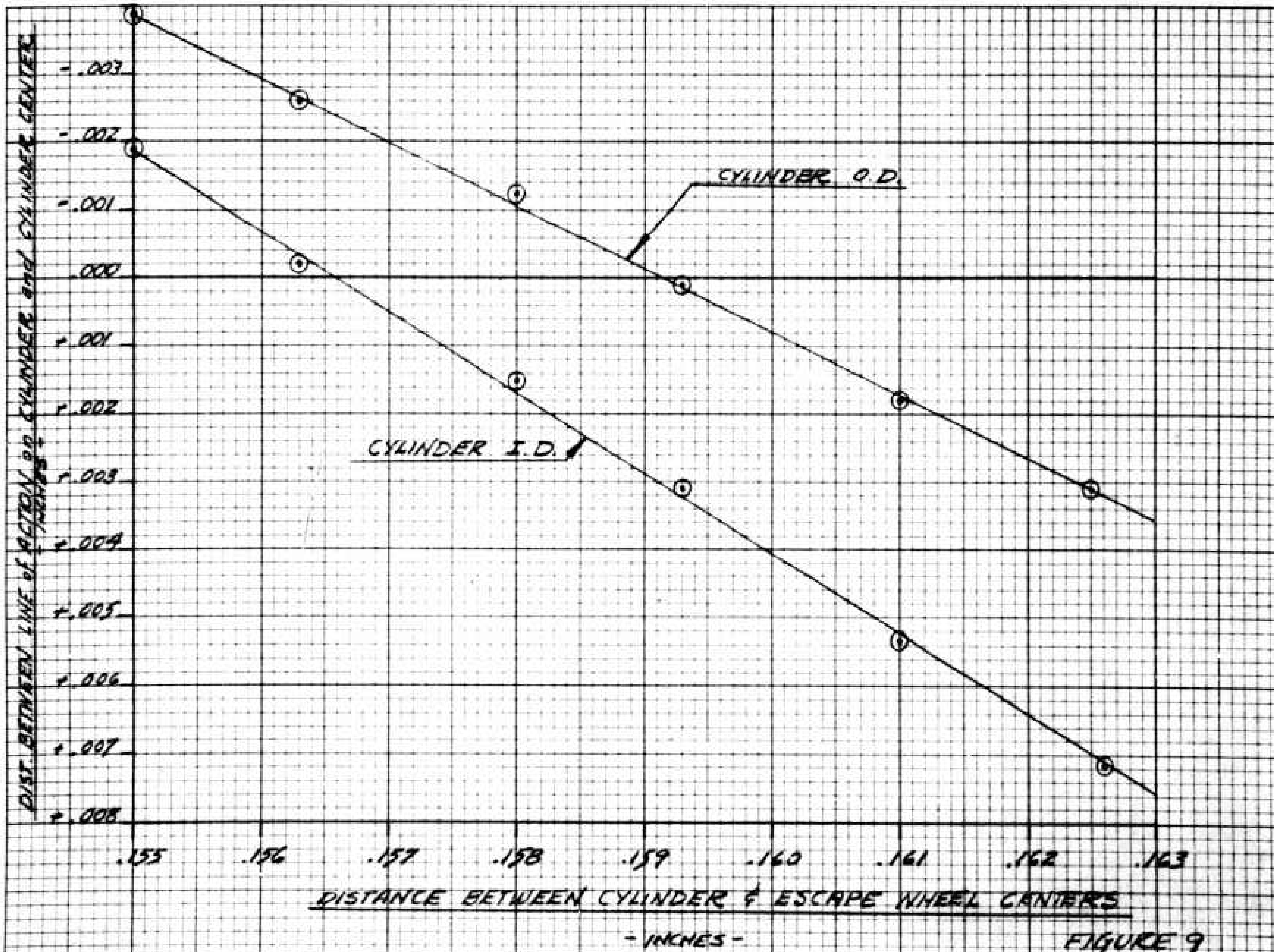


FIGURE 9

1582-D-12 9-30-61

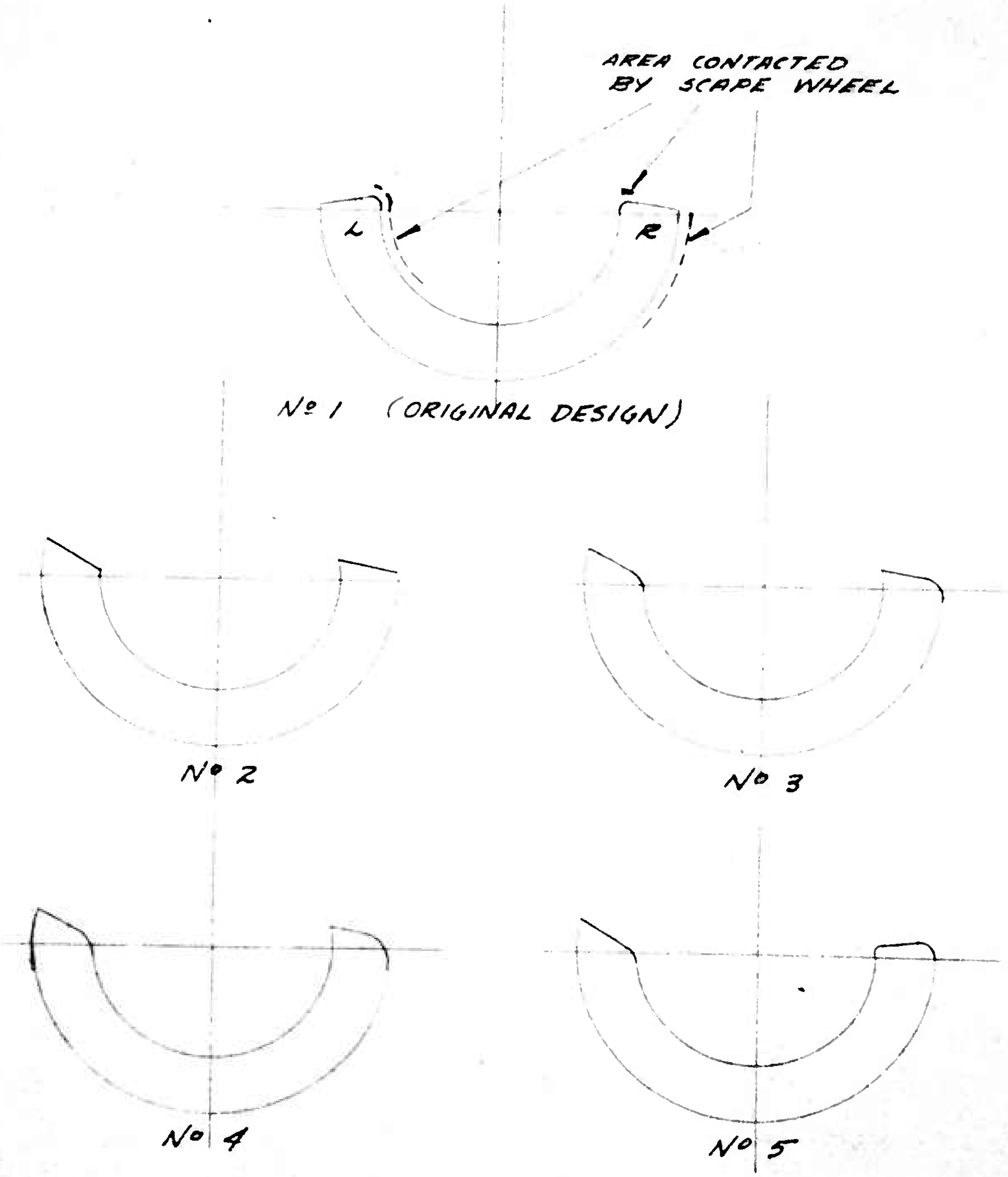
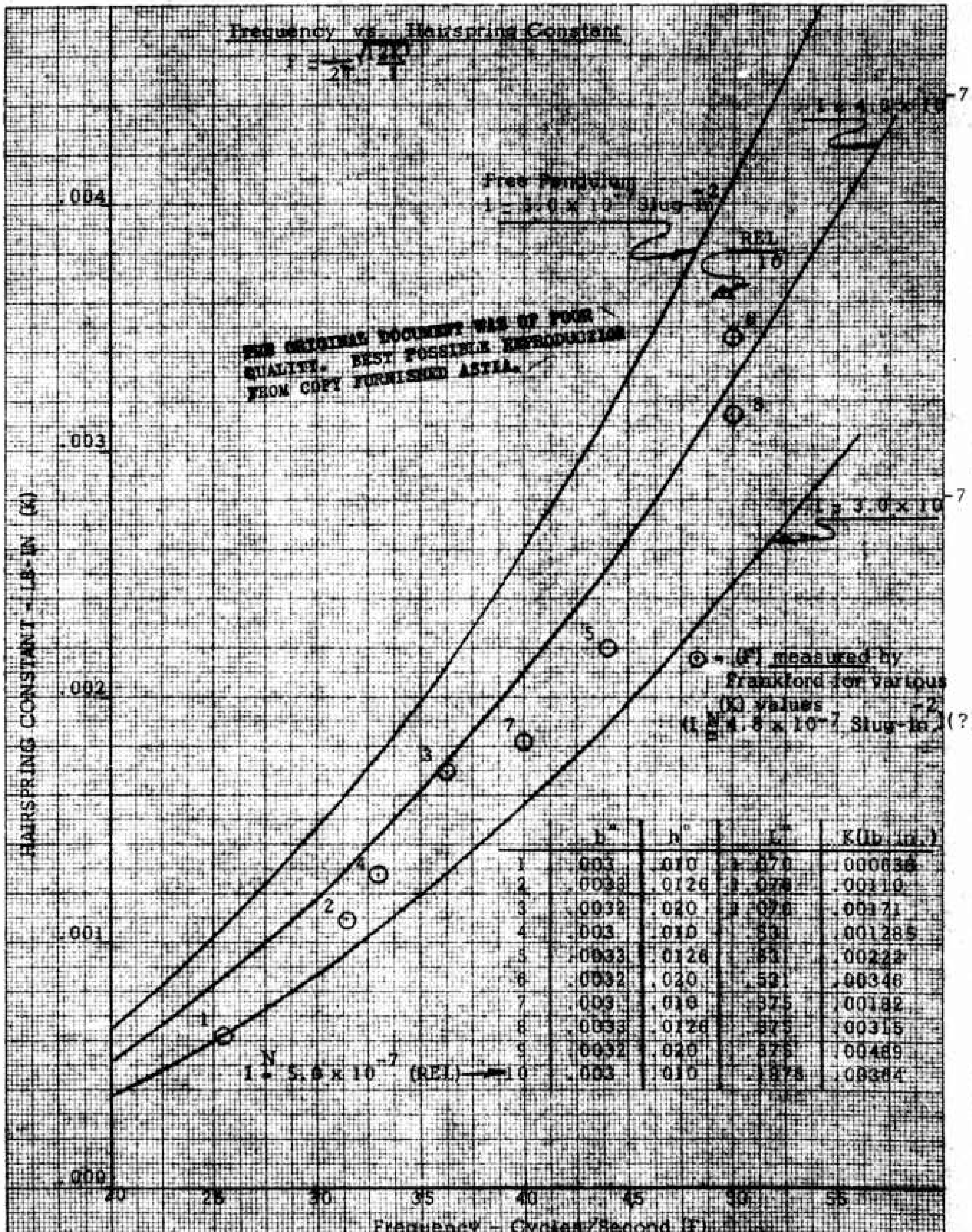


FIGURE 10
CYLINDER PALLET GEOMETRY 30:1
CMS 7-21-61



	b"	h"	L"	K(lb in.)
1	.003	.010	1.070	.000638
2	.0033	.0126	1.070	.00110
3	.0032	.020	1.070	.00371
4	.003	.010	.831	.001285
5	.0033	.0126	.831	.00222
6	.0032	.020	.521	.00346
7	.003	.010	.875	.00182
8	.0033	.0126	.875	.00315
9	.0032	.020	.875	.00489
10	.003	.010	.875	.00364

Figure II

359-14 KEUFFEL & ESSER CO. Millimeters, 5 mm. lines accented, cm. lines heavy. MADE IN U.S.A.

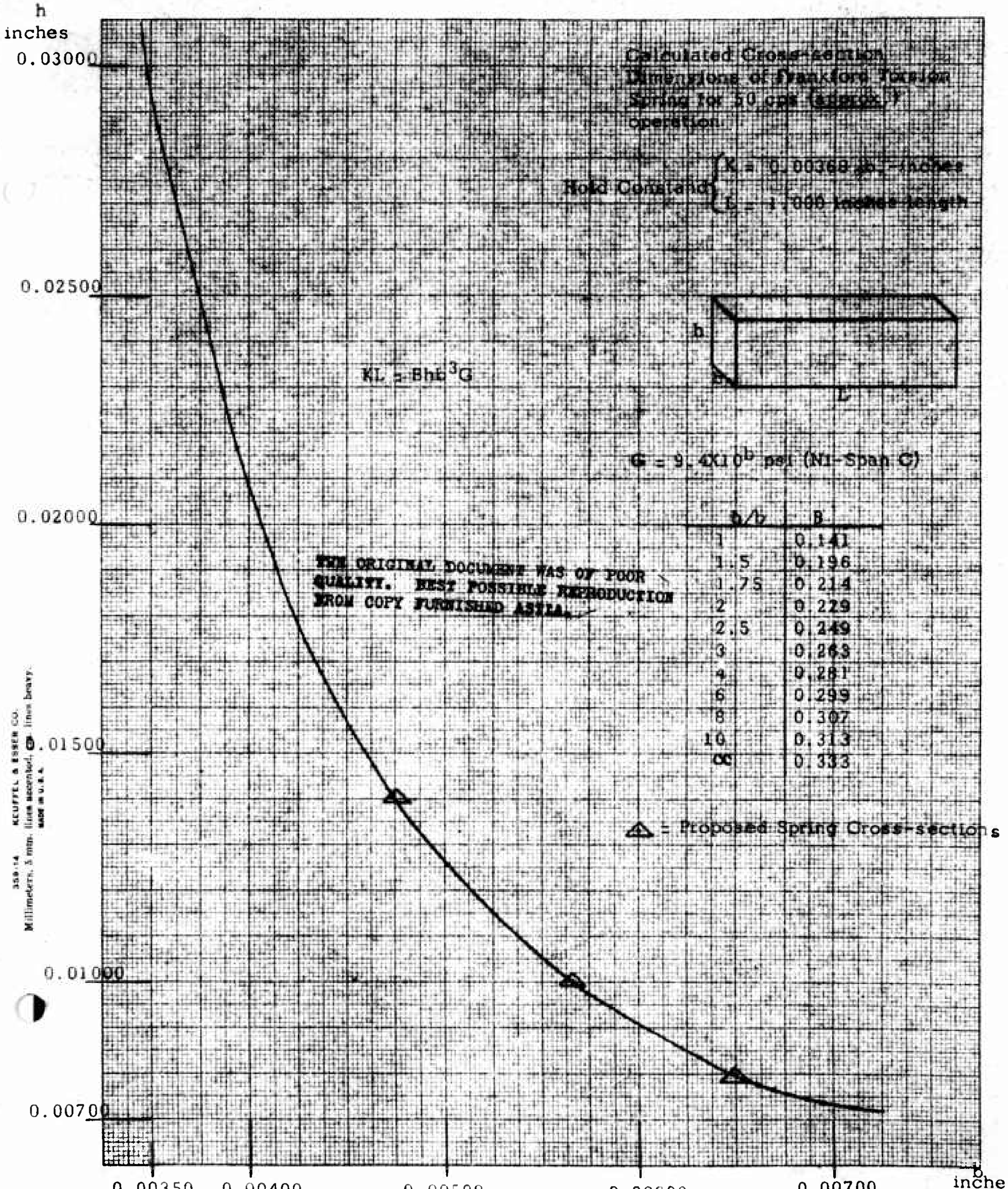


Figure 12

1582-D-9

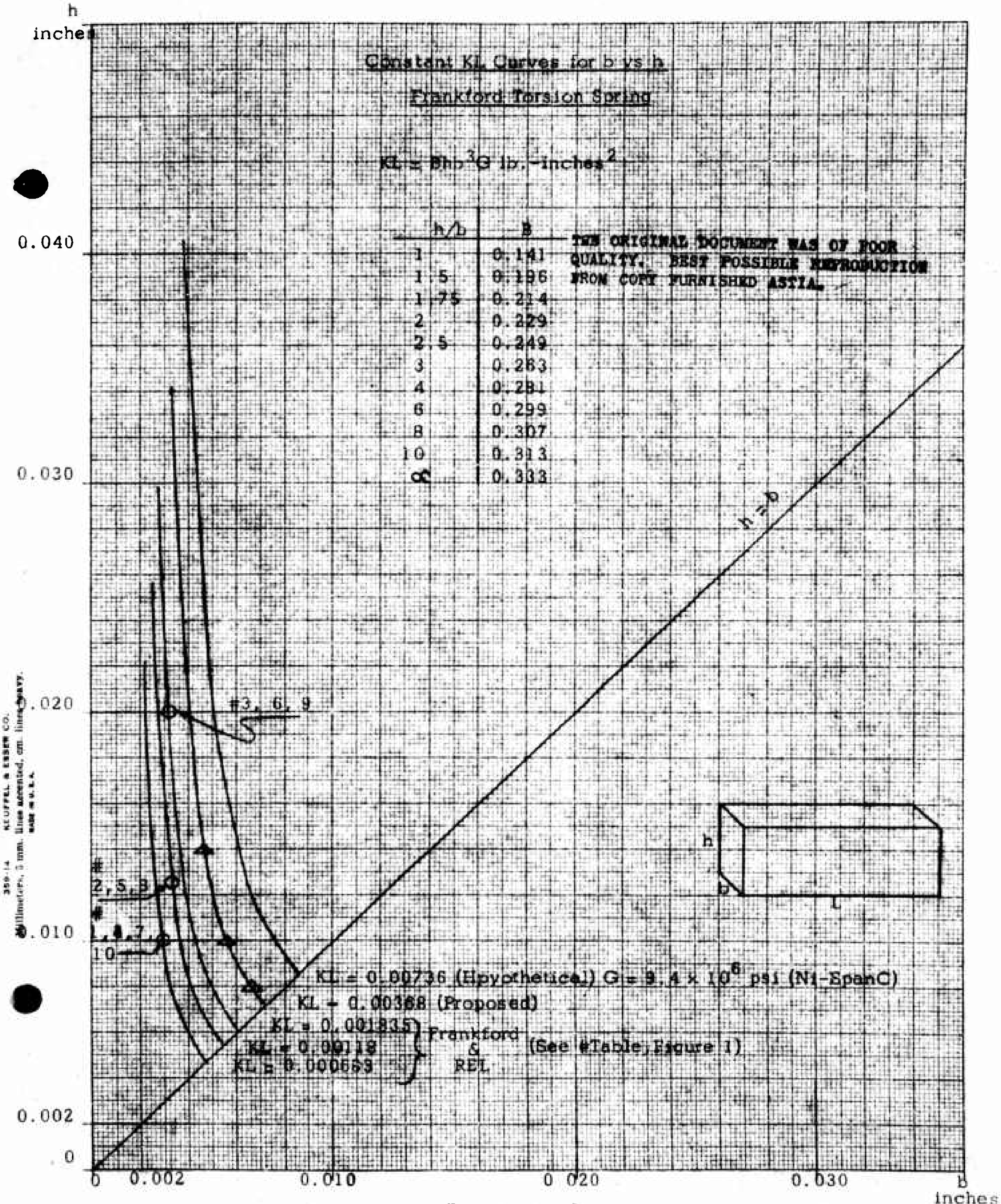


Figure 13

1582-D-10

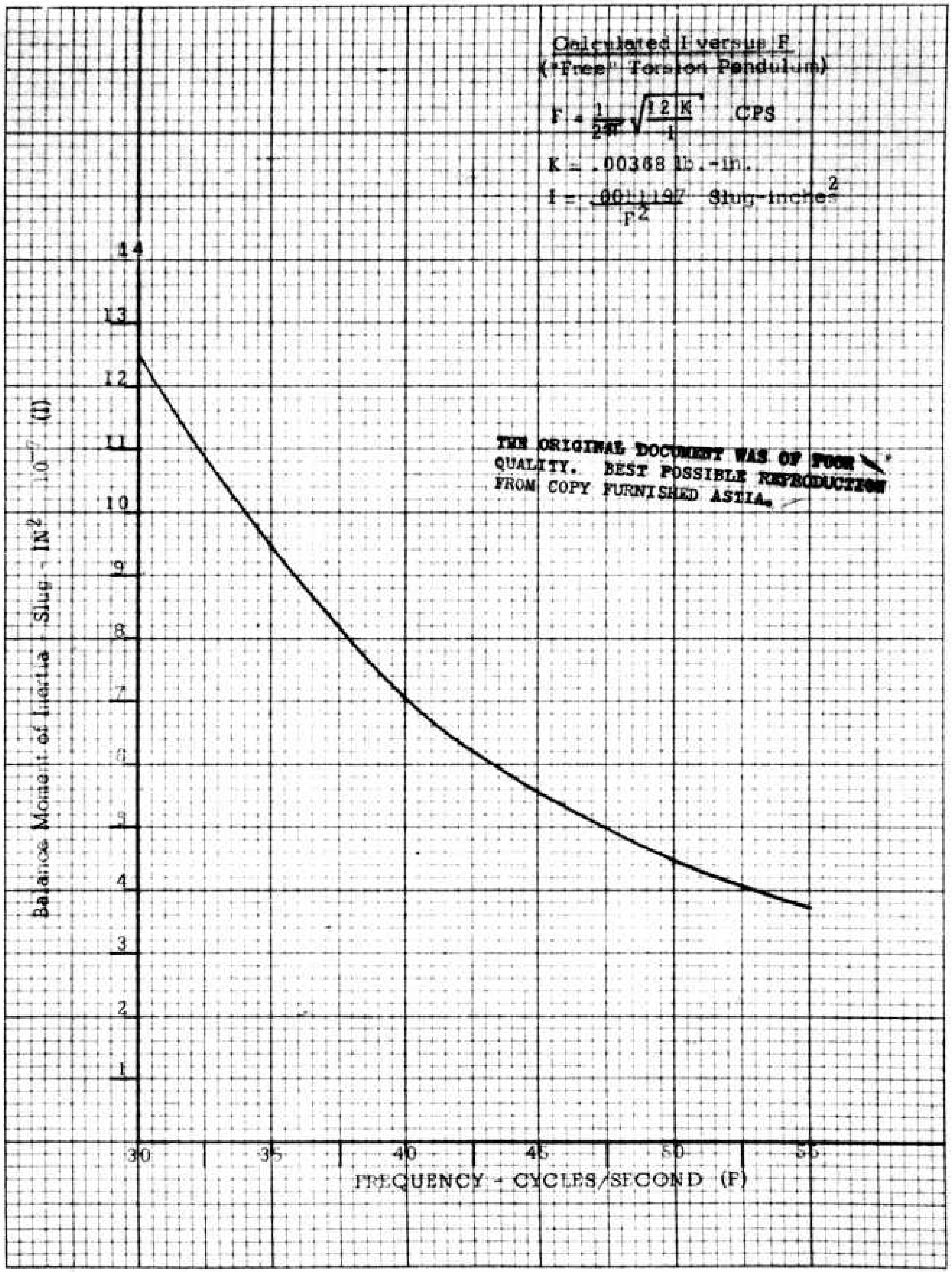


Figure 14

1582-D-11

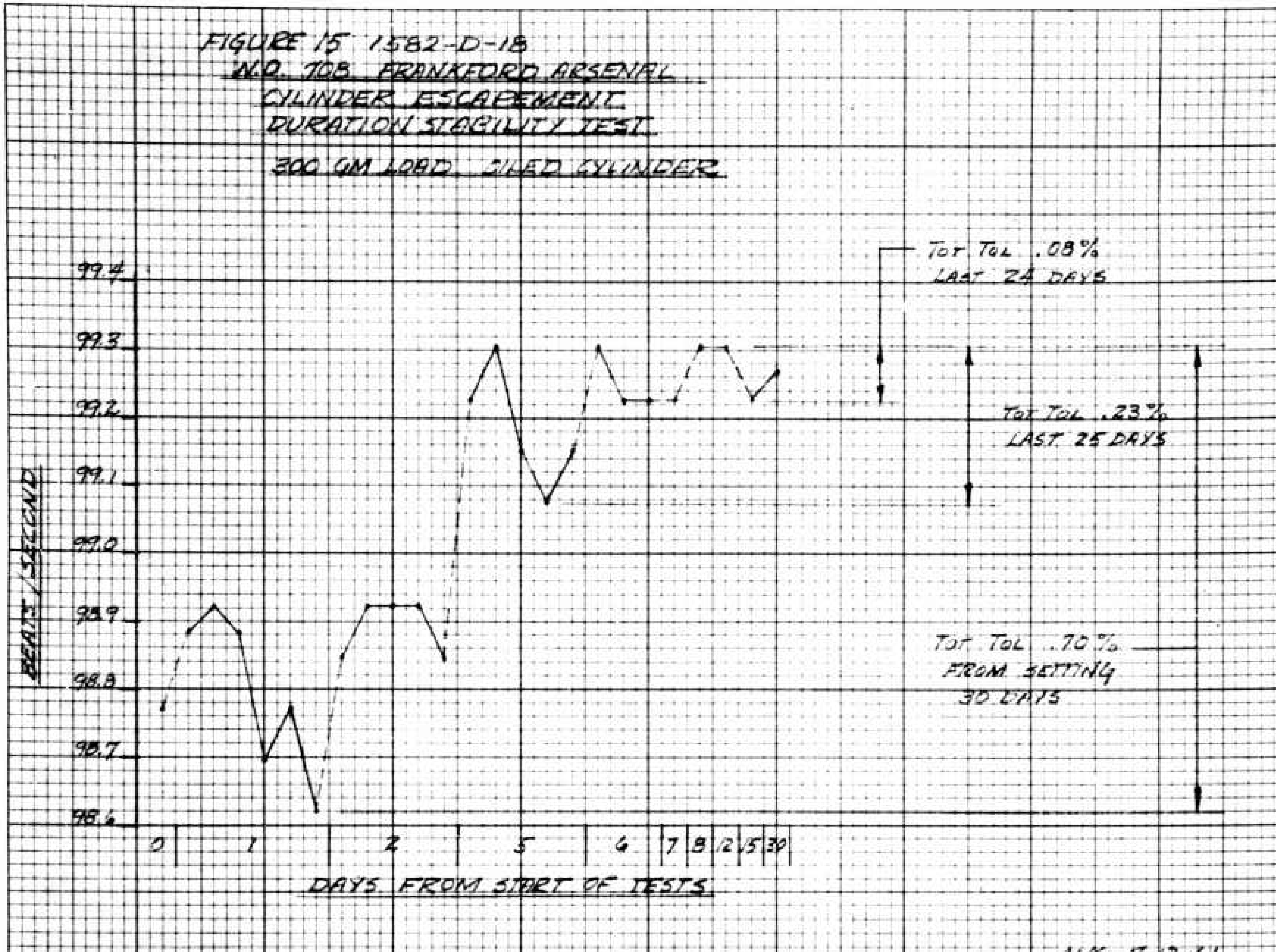


FIGURE 15 1582-D-18
W.O. 708 FRANKFORD ARSENAL
CYLINDER ESCAPEMENT
DURATION STABILITY TEST
300 GM LOAD OILED CYLINDER
BEATS / SECOND
99.4
99.3
99.2
99.1
99.0
98.9
98.8
98.7
98.6
0 1 2 5 6 7 8 12 15 30
DAYS FROM START OF TESTS
TOT TOL .08%
LAST 24 DAYS
TOT TOL .23%
LAST 25 DAYS
TOT TOL .70%
FROM SETTING
30 DAYS
CMS 7-12-61

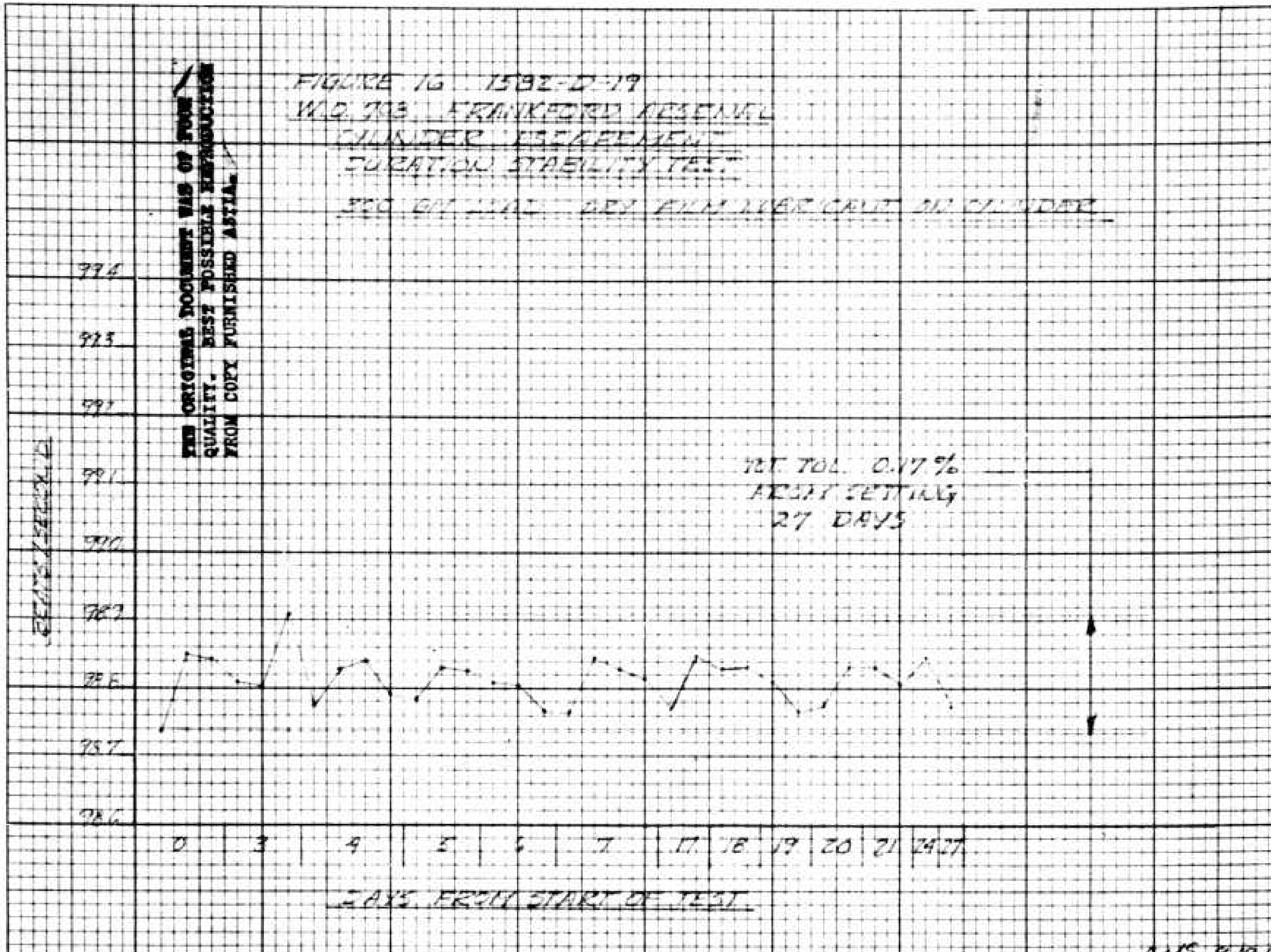
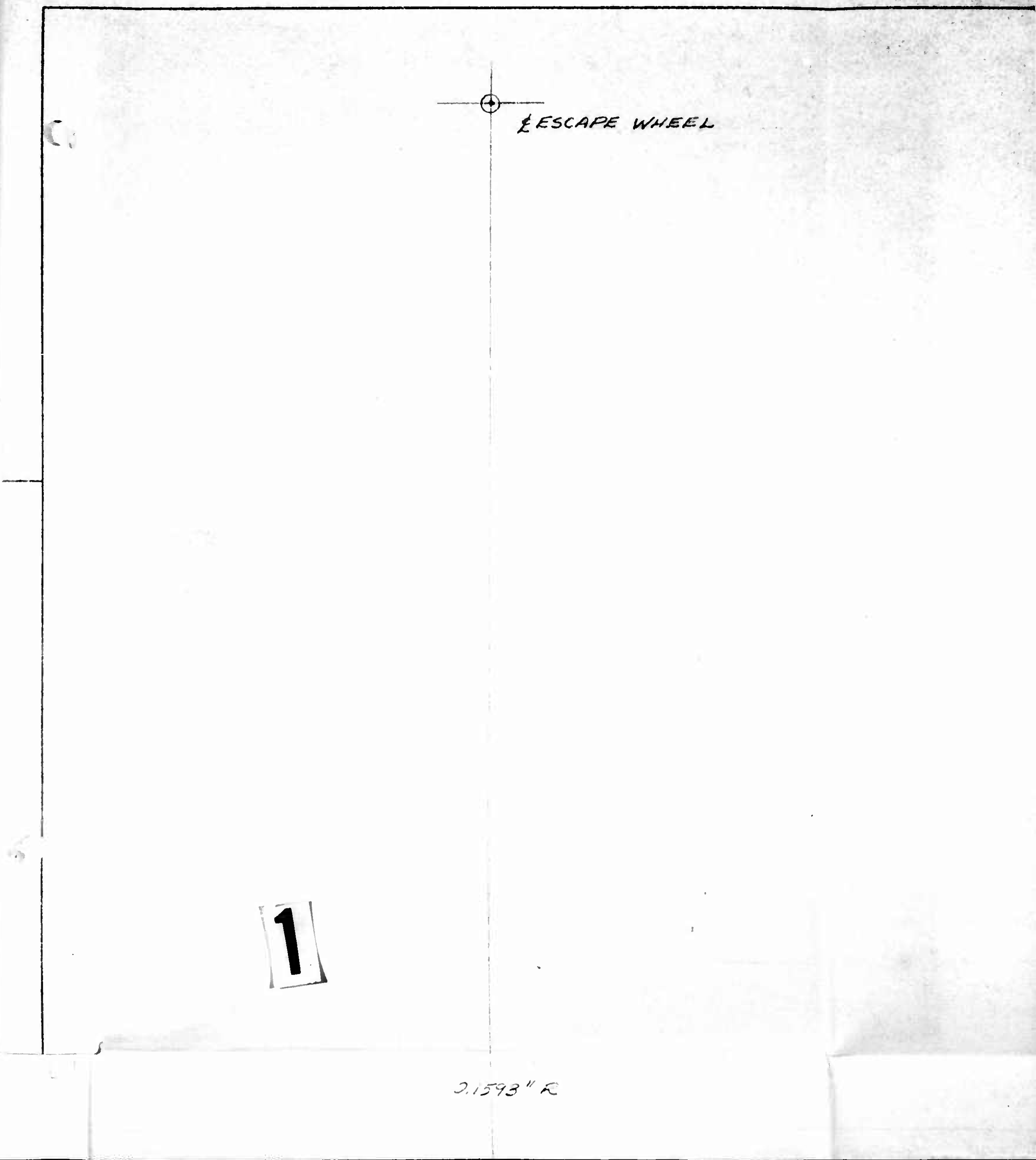
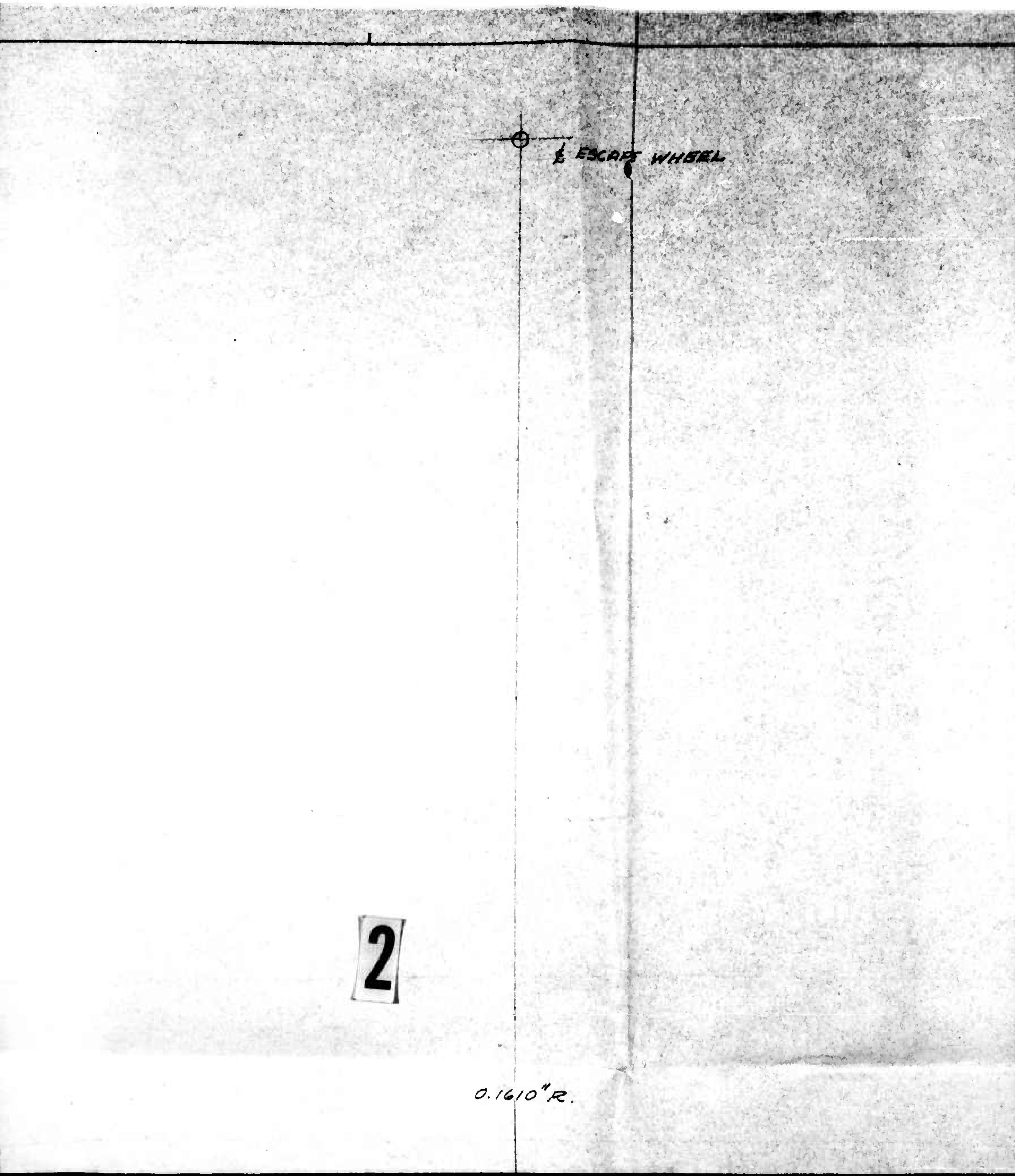
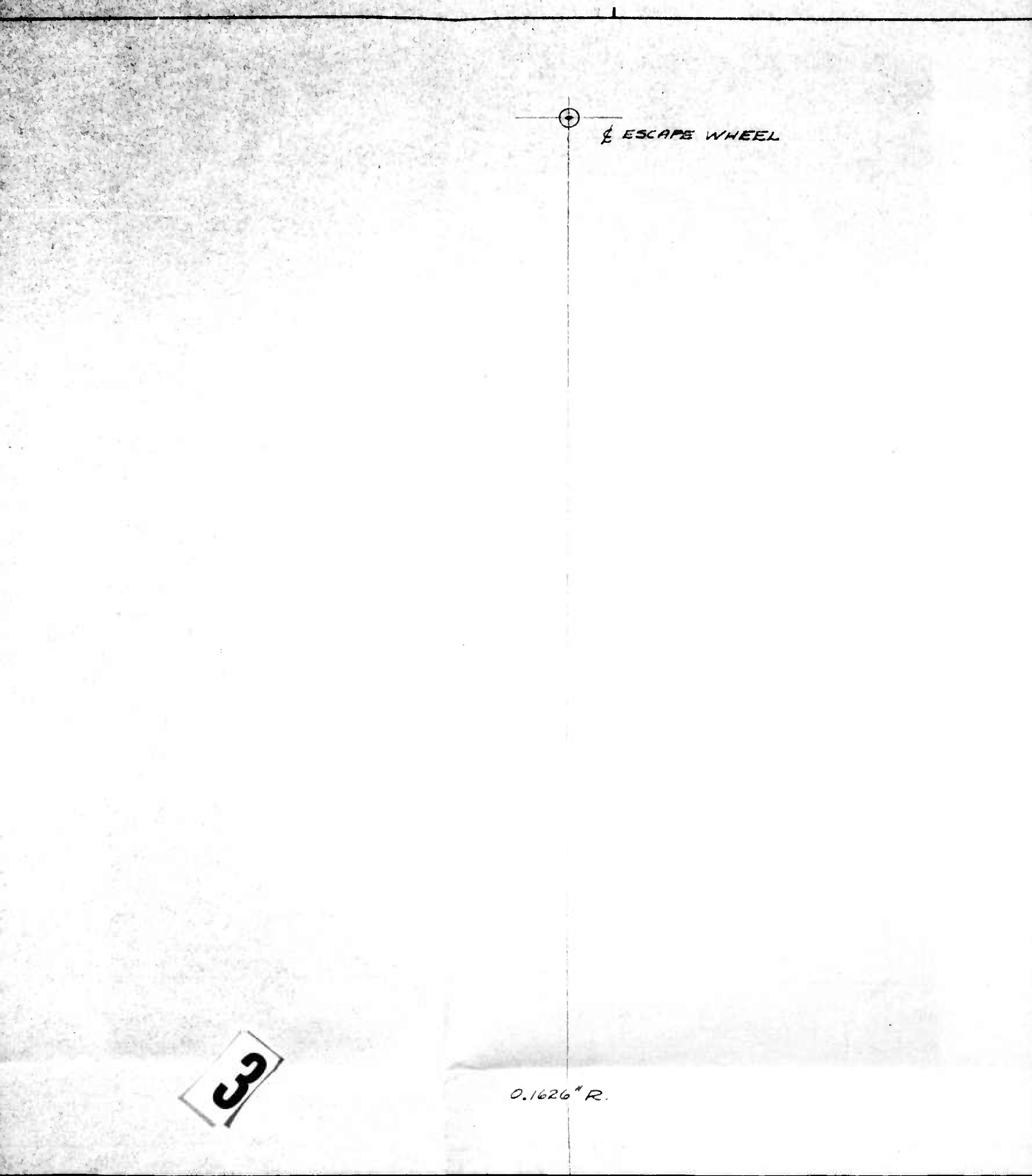


FIGURE 16 1582-D-17
W.O. 703 FRANKFORD ARSENAL
CYLINDER ESCAPEMENT
DURATION STABILITY TEST

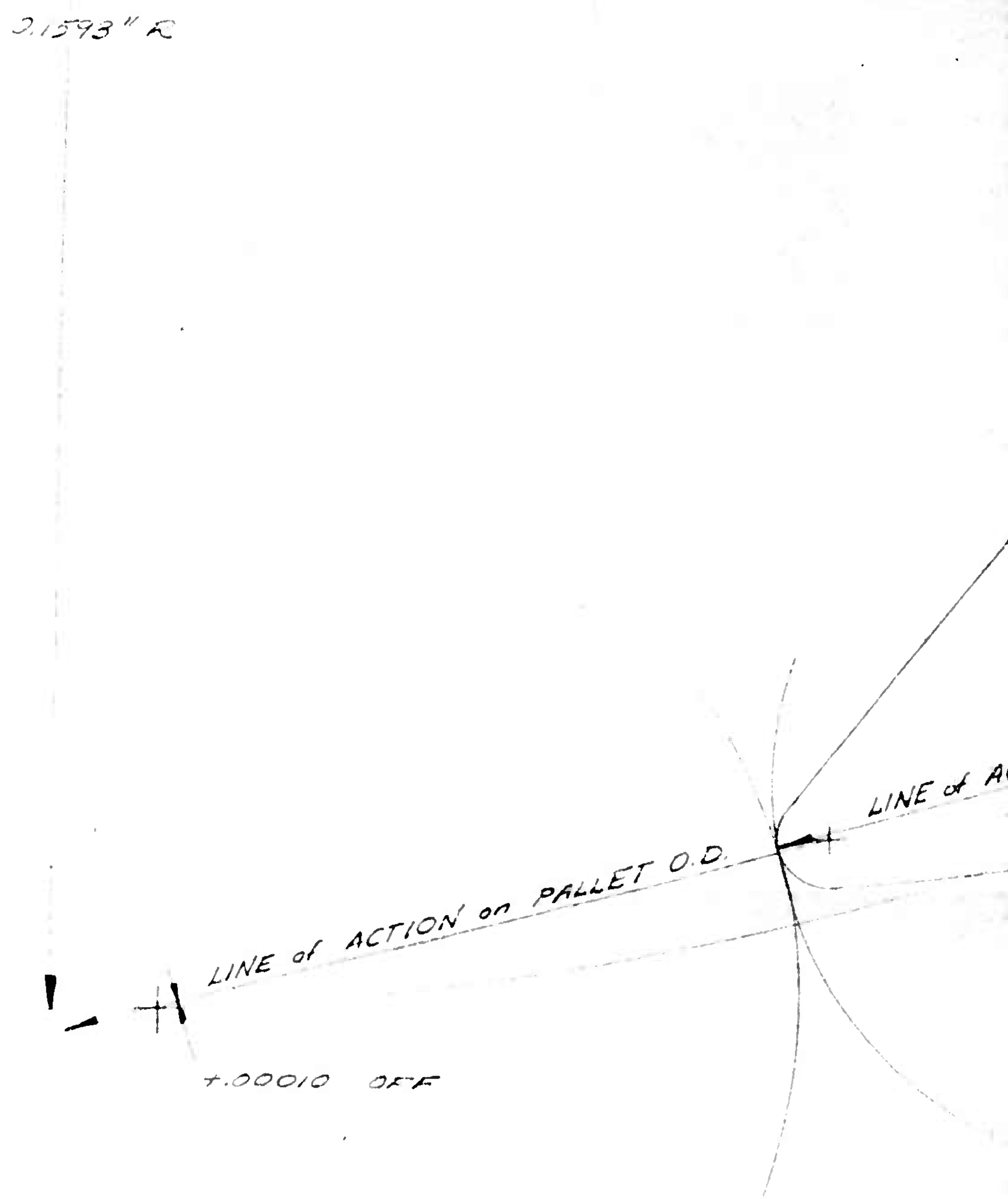

℄ ESCAPE WHEEL
1
0.1593" R


℄ ESCAPE WHEEL
0.1610" R.
2


℄ ESCAPE WHEEL
0.1626" R.
3

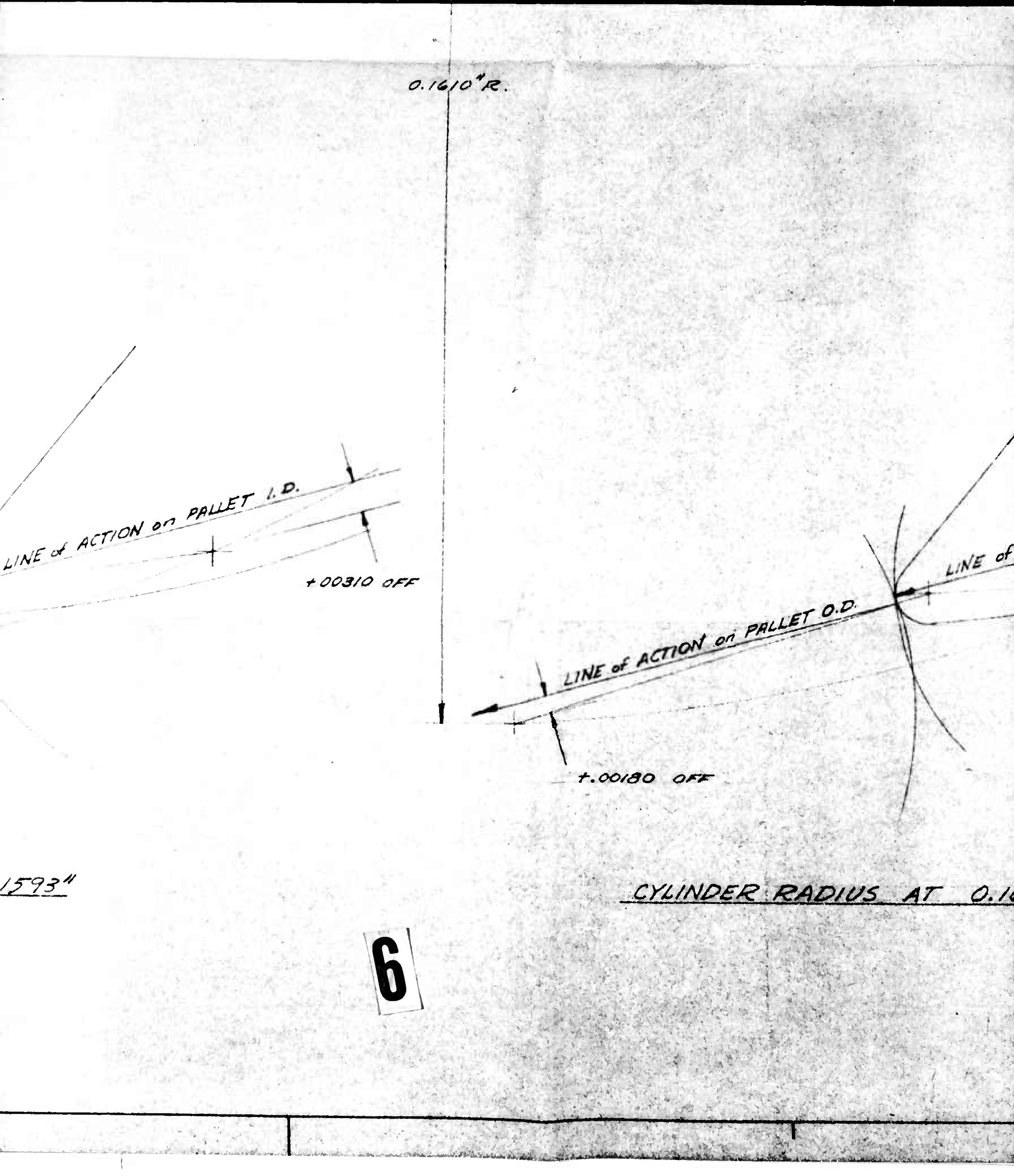
REVISIONS				DWG. NO. 1582-4	REV.
SYM.	DESCRIPTION	BY	DATE		

4

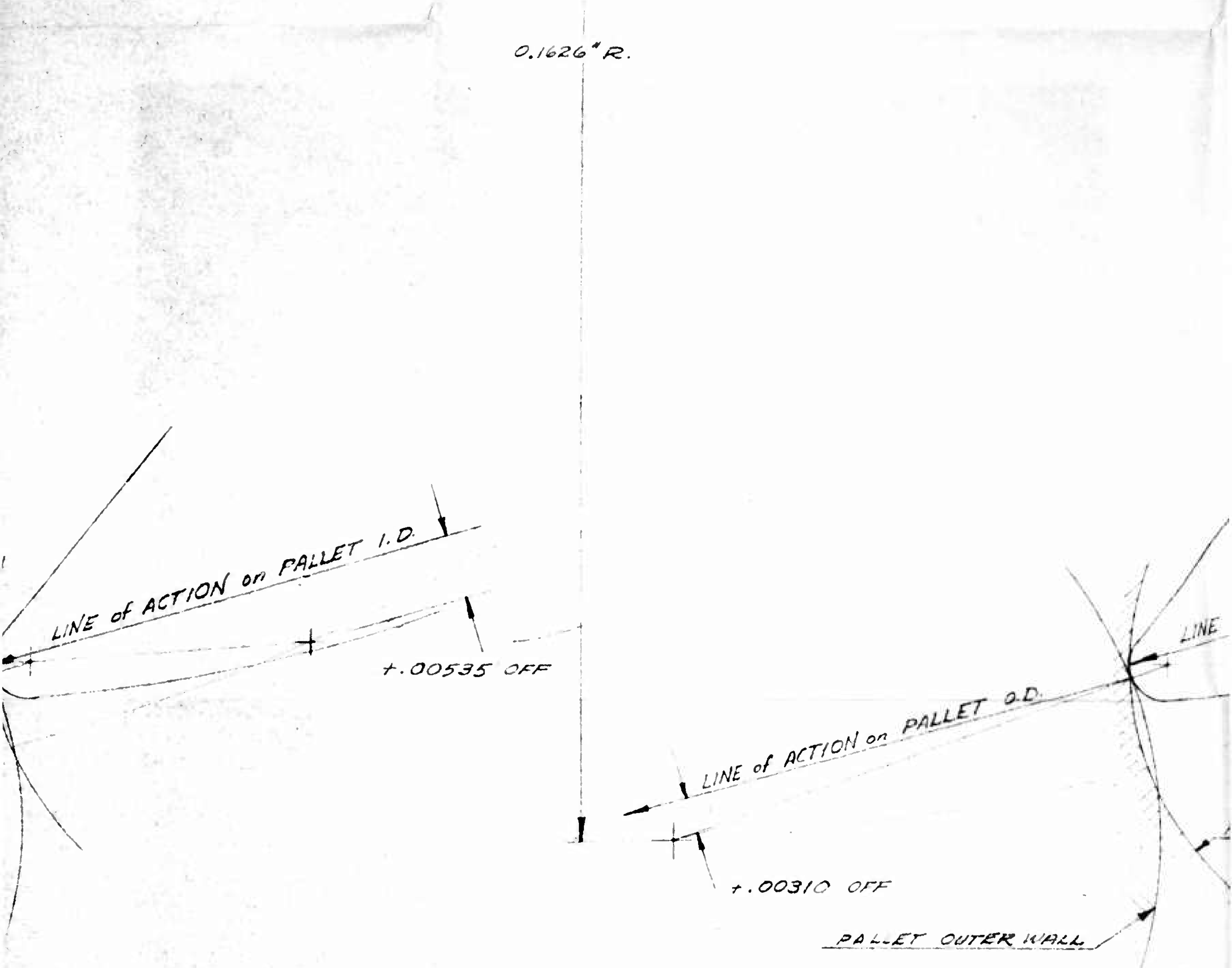


5

CYLINDER RADIUS AT 0.1593"



0.1610" R.
LINE of ACTION on PALLET I.D.
+.00310 OFF
LINE of
LINE of ACTION on PALLET O.D.
+.00180 OFF
1593"
CYLINDER RADIUS AT 0.1
6



AT C.1610"

CYLINDER RADI

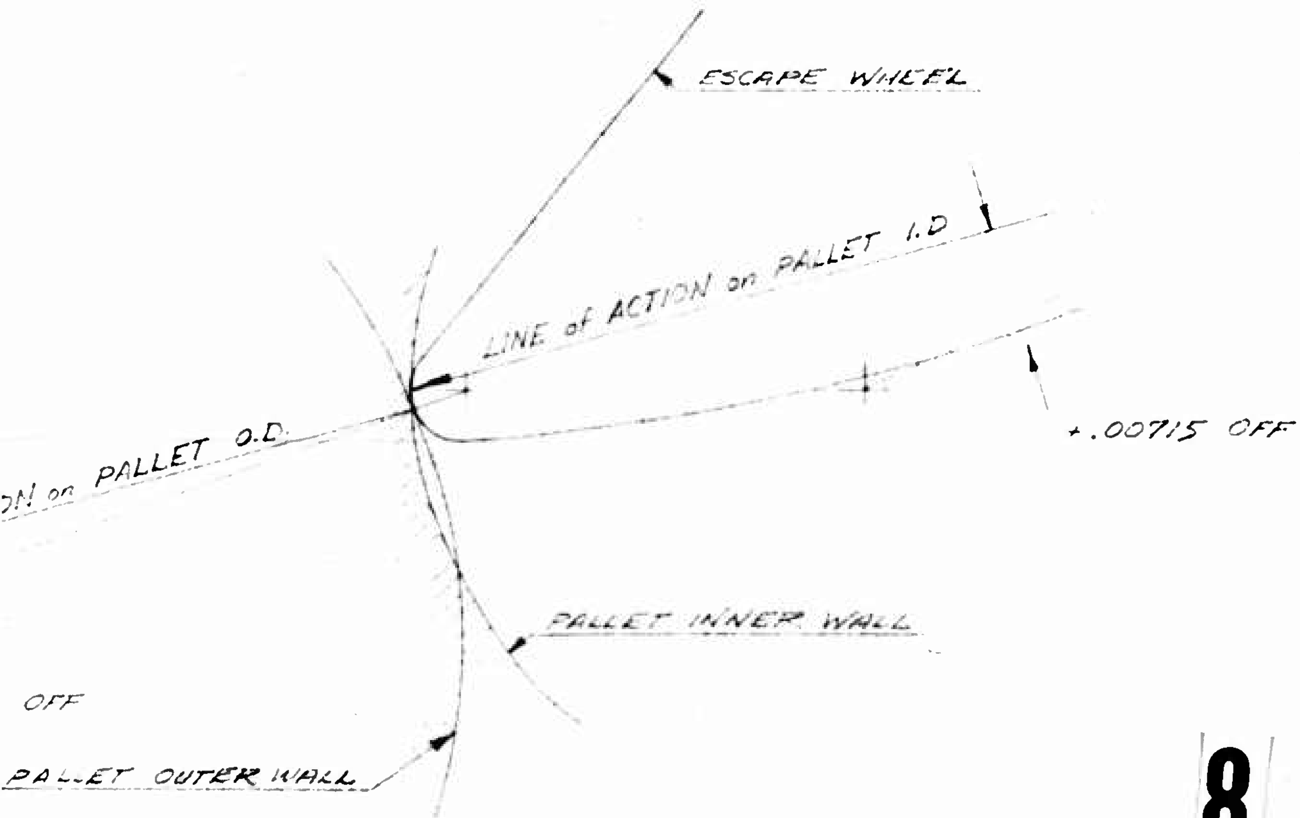
1

LIMITS OF ACCEPTABLE WORKMANSHIP ARE DEFINED IN MP-2000

TOLERANCES (UNLESS OTHERWISE SPECIFIED)		NEXT ASS'Y.	NO. REQ'D.	RAYMOND ENGI MIDDLET
DECIMAL DIM.	± .005			
FRACTIONAL DIM.	± 1/64			MATERIAL
ANGULAR DIM.	± 5°			SPEC.
ALL DIMENSIONS GIVEN IN INCHES				

DWG. NO. 1582-4

REV.



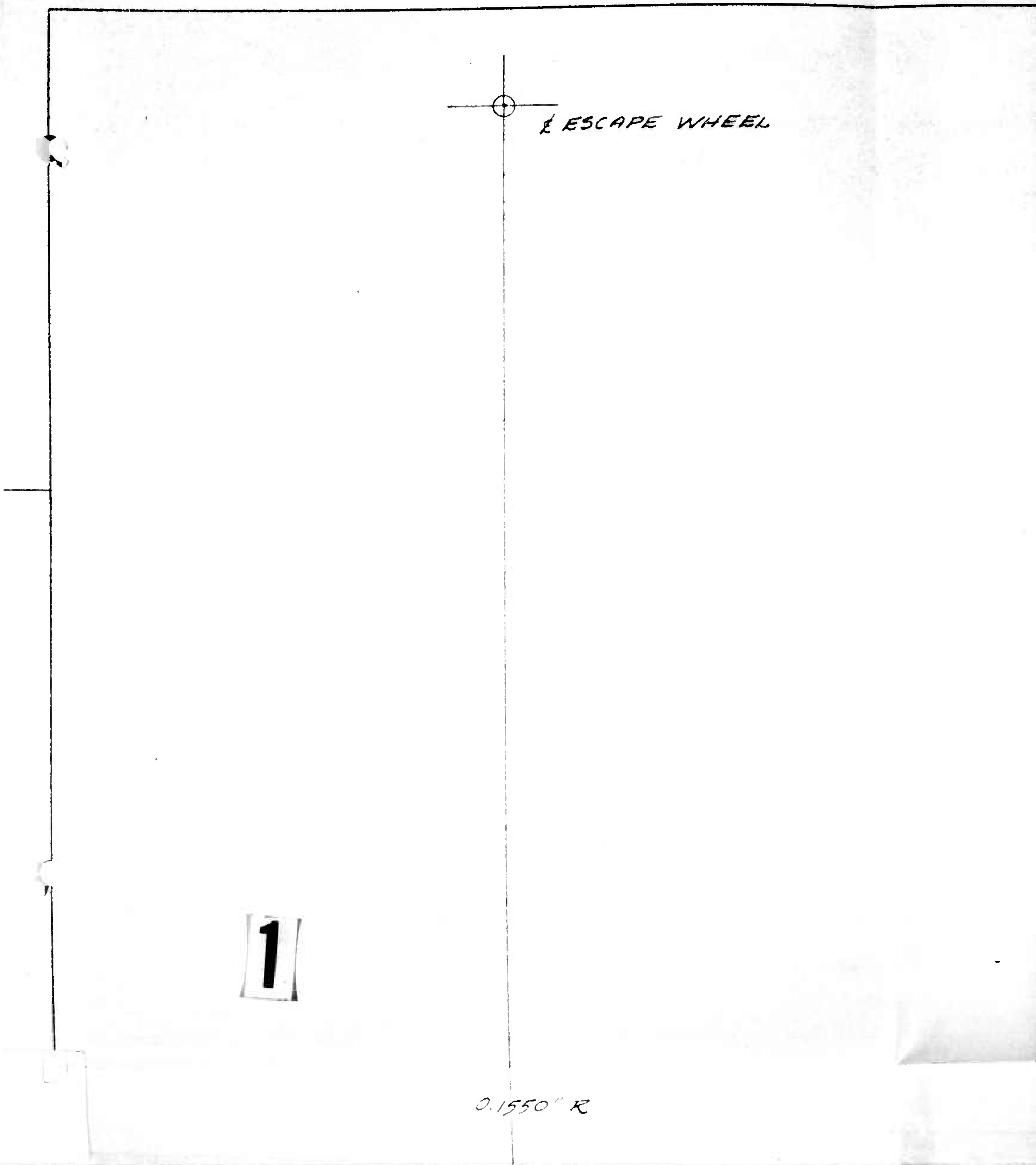
8

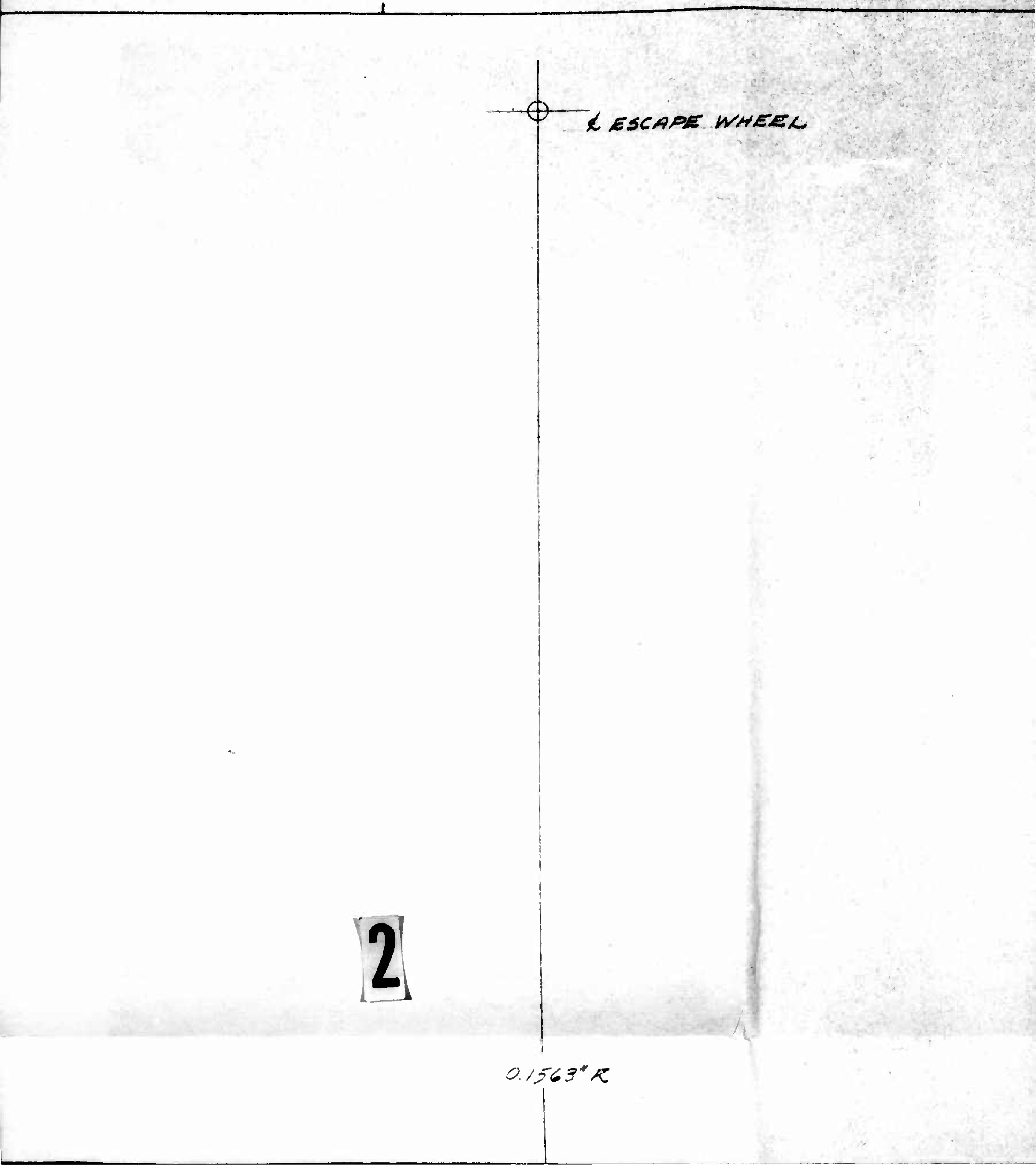
CYLINDER RADIUS AT 0.1626"

ORKMANSHIP ARE DEFINED IN MP-2000

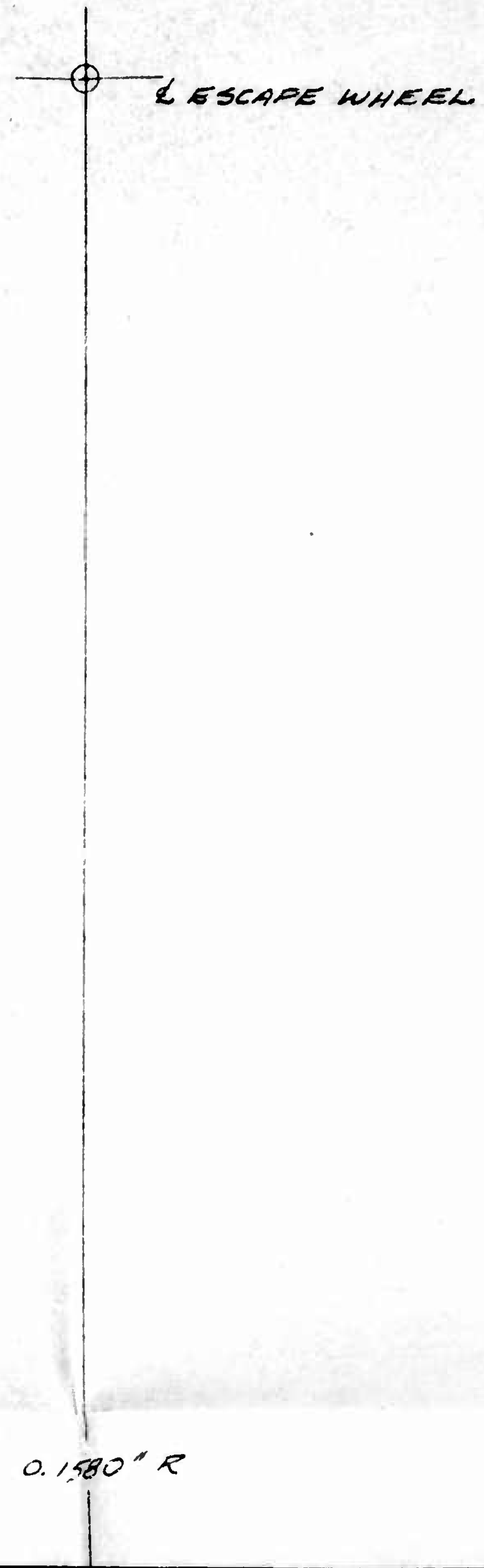
ES SPECIFIED) ± .005 ± 1/64 ± 5° NS CHES	NEXT ASS'Y.	NO. REQ'D.	RAYMOND ENGINEERING LABORATORY, INC. MIDDLETOWN, CONNECTICUT		W.O. 708 NO. REQ'D.	TITLE ℄ DIST. STUDY		
			MATERIAL	FINISH	DRAWN CMS	DATE 3-29-61	SCALE 100 : 1	
			SPEC. ——	SPEC. ——	CHECKED	DATE	DWG. SIZE D	DWG. NO. 1582-4	REV.
					APPROVED	DATE		

PRINT ISSUED AUG 10


℄ ESCAPE WHEEL
1
0.1550" R



2



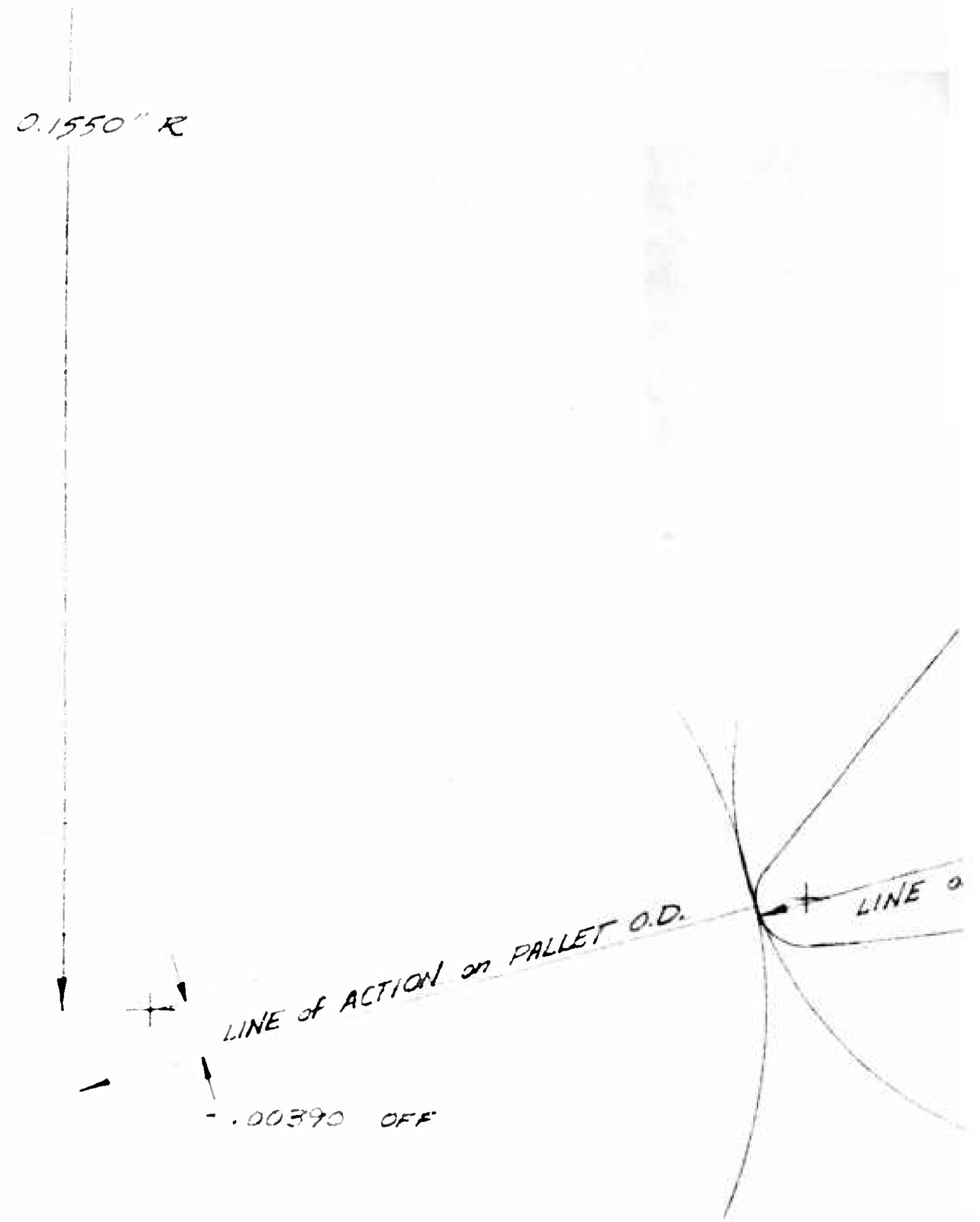
3

HEEL.

REVISIONS	DWG. NO. 1582-5	REV.

SYM.	DESCRIPTION	BY	DATE

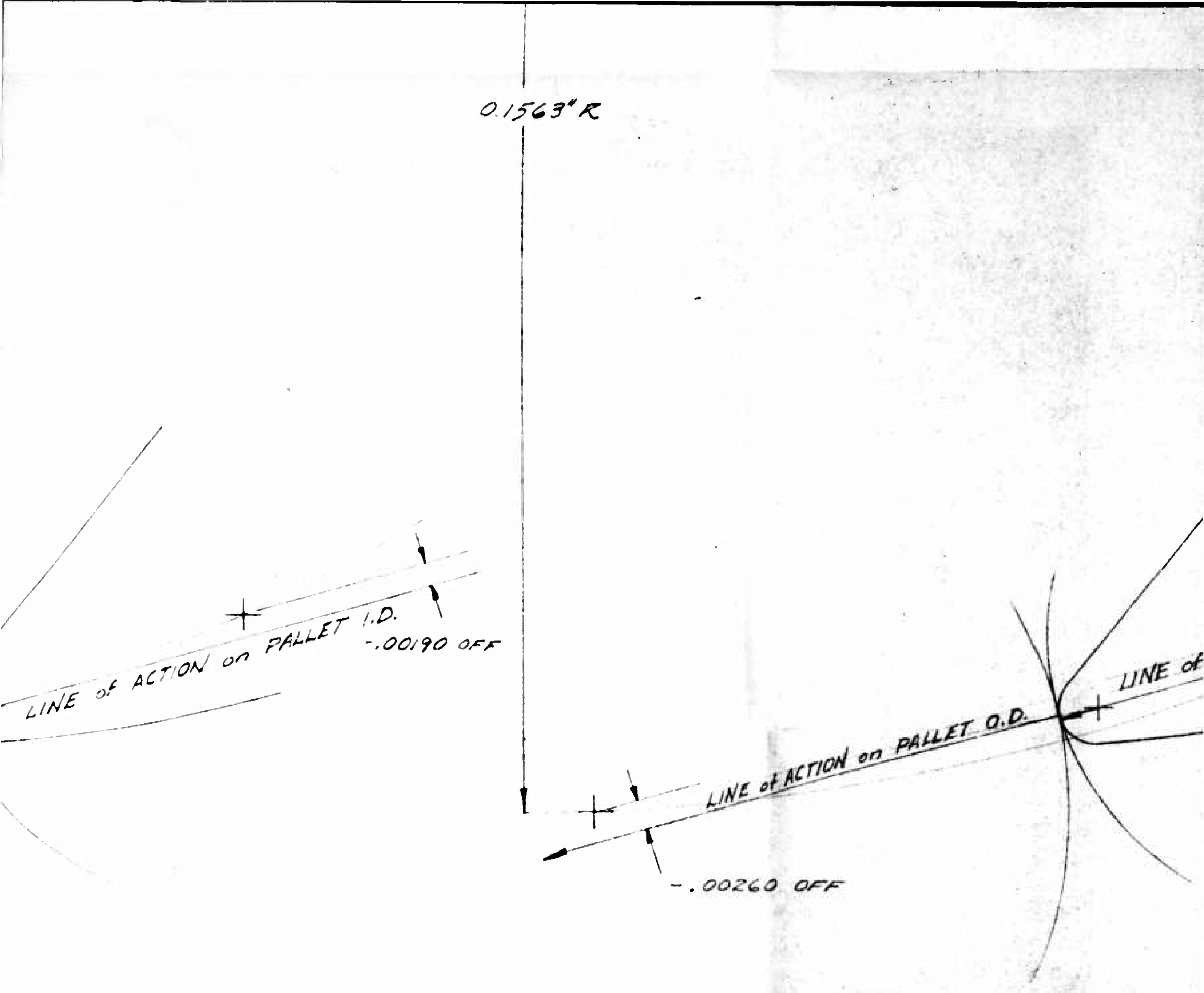
4



CYLINDER RADIUS AT 0.1550"

5

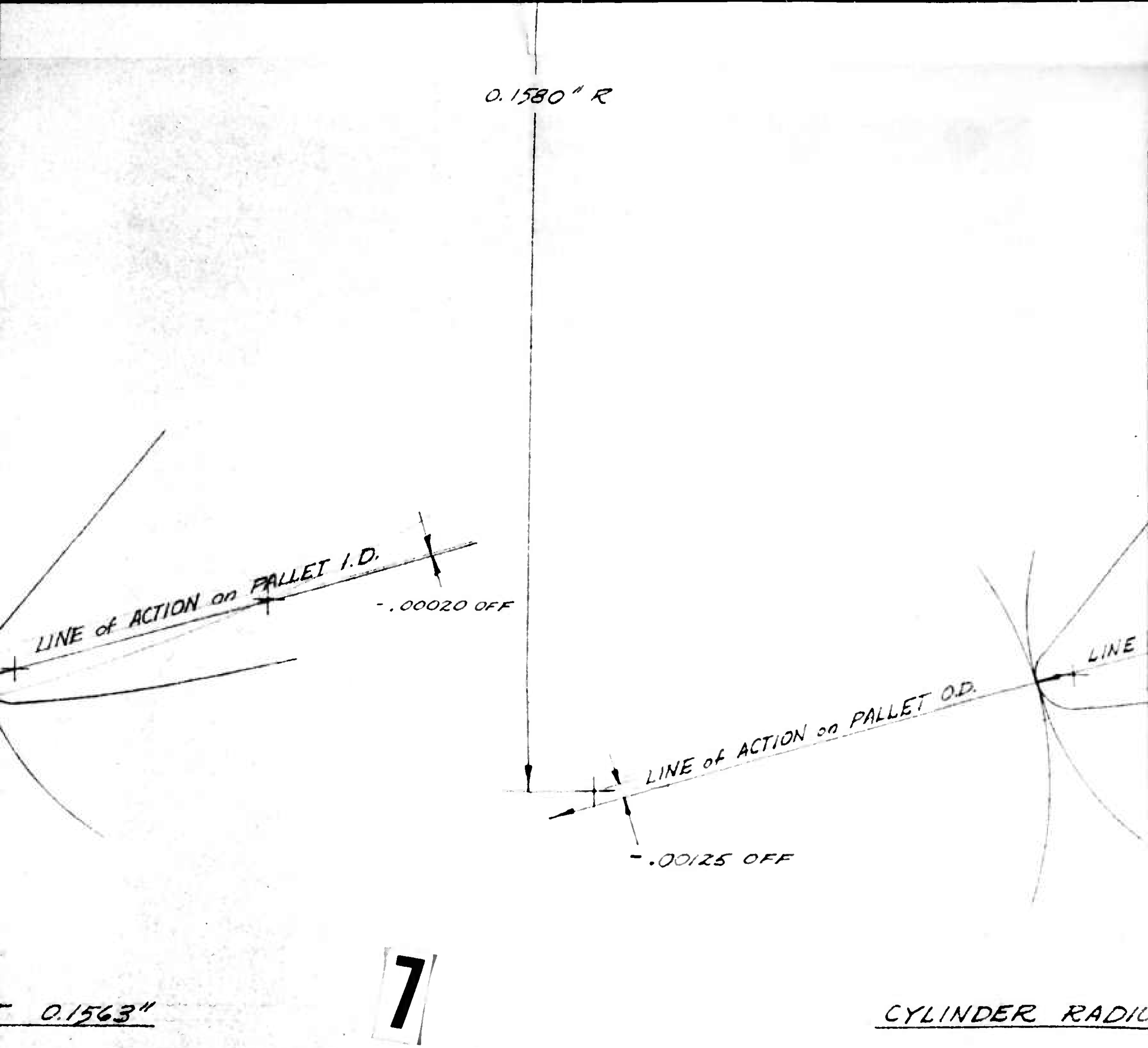
IINT 1000H



.1550"

6

CYLINDER RADIUS AT 0.156



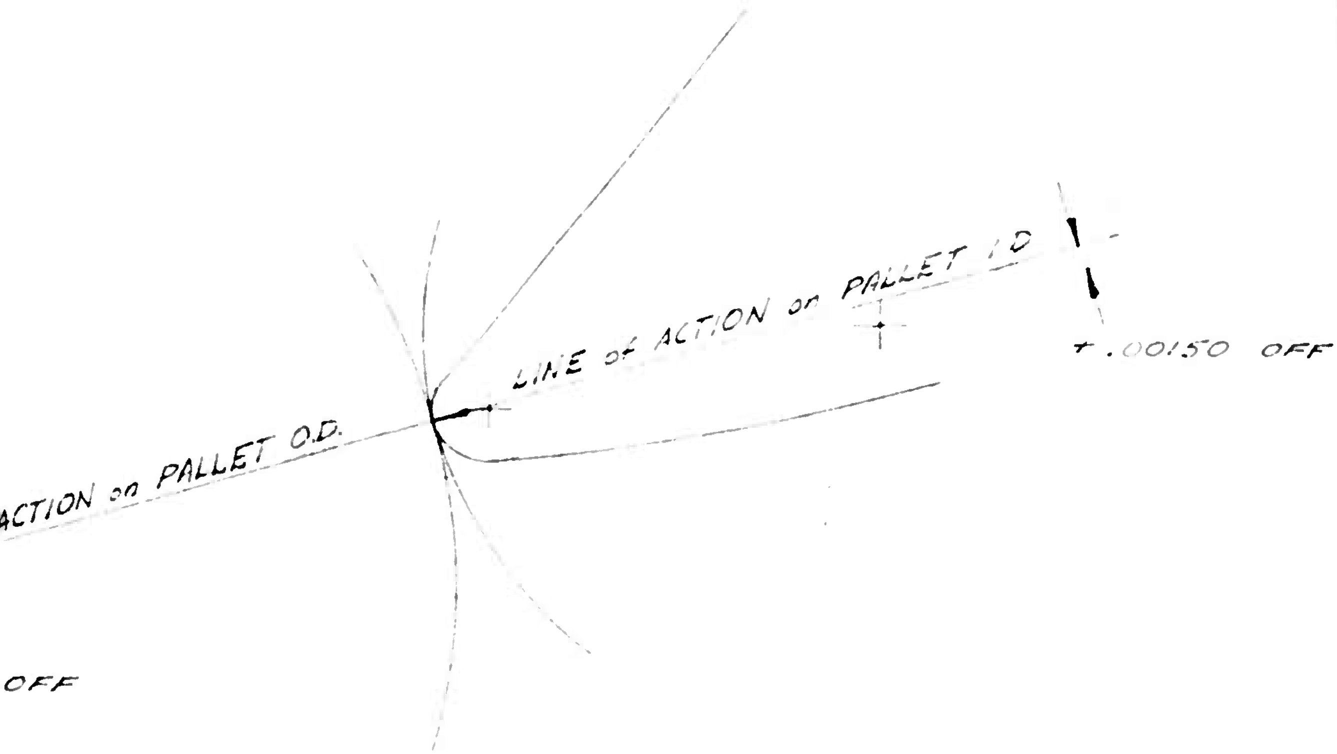
7

LIMITS OF ACCEPTABLE WORKMANSHIP ARE DEFINED IN MP-2000

TOLERANCES (UNLESS OTHERWISE SPECIFIED)	NEXT ASS'Y.	NO. REQ'D.	RAYMOND ENGIN
DECIMAL DIM. ± .005			MIDDLETO
FRACTIONAL DIM. ± 1/64			MATERIAL
ANGULAR DIM. ± .5°			SPEC.
ALL DIMENSIONS GIVEN IN INCHES			

DWG NO. 1582-5

REV.



CYLINDER RADIUS AT 0.1580"

8

WORKMANSHIP ARE DEFINED IN MP-2000

ICES (…E SPECIFIED) ± .005 …M. ± 1/64 ± .5° …SIONS … INCHES	NEXT ASS'Y.	NO. REQ'D.	RAYMOND ENGINEERING LABORATORY, INC. MIDDLETOWN, CONNECTICUT		W.O. 708 NO. REQ'D.	TITLE ℄ DIST. STUDY	
			MATERIAL SPEC.	FINISH SPEC.	DRAWN CMS	DATE 3-29-61	SCALE 100:1
					CHECKED	DATE	DWG. SIZE D / DWG NO. 1582-5 / REV.
					APPROVED	DATE	

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